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Photo by] SIR WILLIAM H. WHITE, K.C.B., F.R.S., &c. *[J. Russell & Sons, Southsea.*

Yours truly

W H White

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Foreword.

THE first Annual International Number of *The Shipbuilder* is published in response to the suggestions of many readers that we should devote more attention than we are able to do in our ordinary numbers to the numerous papers communicated to the British and foreign scientific and technical institutions which are of value to those engaged in shipbuilding and marine engineering. This is the first attempt to embody in one volume a world's survey, at once concise and comprehensive, of the scientific progress made each year in the great industry which our journal represents, and we venture to express the hope that it will prove a useful work of reference, particularly to those readers who are chiefly concerned with the more technical phases of the industry.

Our decision to devote considerable attention to the papers read before the foreign societies has been guided by the fact that many British readers are unable to study in the original the papers presented to, say, the German, French, and Italian institutions, which, as contributions to the technical literature of the industry, are yearly becoming more valuable. It is also hoped that the work will provide for our readers abroad a handy *résumé* of the latest developments in naval architecture and marine engineering in the United Kingdom as well as in all foreign centres.

We desire to take this opportunity to place on record our grateful thanks to the distinguished writers of papers in Great Britain, Europe, America, and Japan, and to the secretaries of the various scientific and technical institutions in all these countries, whose courteous assistance has enabled us to publish this volume.

The Design of Ships.

THE MAXIMUM DIMENSIONS OF SHIPS.

By Sir WILLIAM H. WHITE, K.C.B., F.R.S., D.Sc.

(*American Society of Naval Architects and Marine Engineers; November*, 1911.)

IN considering this subject, the chief needs, according to Sir William White, are reasoned replies to certain questions :—

(1) Is it probable that the dimensions of ships will continue to grow at the rapid rate which has prevailed in recent years ?

(2) Will an upper limit to the sizes of ships be reached ?

(3) What considerations, if any, are likely to determine such a limit, for either merchant ships or warships ?

In the author's opinion, certain facts and principles will for the most part command general assent.

I. The law of growth in dimensions has operated hitherto on all classes of ships, and its action has on the whole been beneficial.

II. It must be noted, however, that—notwithstanding the remarkable developments of the last ten years—the number of ships of extremely large dimensions is relatively few. Whatever may be the future growth in size of the largest ships built for special services, the bulk of the maritime business of the world will always be done by ships of relatively moderate dimensions, and they will continue to be the largest contribution to the revenues of port and harbour authorities.

III. Naval architects will agree that, provided the money is forthcoming for building still larger ships, their construction will be possible, even if the materials for shipbuilding already available are not improved upon.

IV. Existing physical conditions in the seaports, harbours, and docks of the world necessarily impose limits on the draught of water of ships.

V. Up to the present time the response made to the appeal from shipowners for increased accommodation for larger ships by authorities of the great seaports of the world has been neither niggardly not unsatisfactory.

VI. The marvellous growth of the world's commerce in modern times has enabled these great engineering works to be undertaken under conditions which have, in most cases, either yielded fair returns to investors, or which have been considered by governments beneficial to the whole nation and worthy of adoption, irrespective of the return on capital expenditure. But while this has hitherto happened, it seems not improbable that a point may be reached beyond which dock and harbour authorities will not care to go.

VII. The dimensions of locks and dock entrances recently constructed or now in process of construction indicate the opinions of responsible authorities in regard to the provision of suitable accommodation for the largest ships likely to be built. On the Panama Canal the locks are 1,000 ft. long, 110ft. wide, with 40ft. of water over the sills. On the re-constructed Kiel Canal the locks are to be 1,080ft. long, 147ft. wide, with 36ft. of water over the sills, but with provision in the structures to increase this to 46ft. if required hereafter. The Gladstone Dock at Liverpool will be 1,020ft. long, 120ft. broad, with 44ft. of water over the sills at high water ordinary spring tides. On the Clyde a dock of nearly the same dimensions is now under construction with 36ft. of water over the sills at high water. The Port of London Authority contemplate the construction of a dock 1,000ft. long, 120ft. wide, with 40ft. of water over the sills. For the new naval base at Rosyth the plans originally provided for a lock 850ft. long, 110ft. wide, with 36ft. of water at low tide, but it is understood that some increase in length and breadth is probable. As to approach channels, a depth of 40ft. at low water is to be secured throughout the Ambrose Channel. At Southampton access is to be secured at all times for ships drawing 34 to 35ft. At Liverpool the minimum depth of water at low tide in the "cut" through the bar of the Mersey is maintained at 30ft. On the St. Lawrence works in hand are intended to give a minimum depth of 35ft. from Montreal to the sea.

VIII. When the foregoing particulars are considered in the light of dimensions of recent large ships, it will be seen that the margins provided are not very great in regard to length

and breadth, although they are expected to prove sufficient for many years to come.

IX. It is a fact worth noting that ships of the maximum dimensions now built or building are not easily manœuvred in the largest docks and harbours. Even in the terminal ports of the Transatlantic steamers, difficulties are necessarily experienced; and although they have been overcome so far, they must be accentuated by any further increase in size.

X. For cargo steamers and warships no such fixed conditions of service or terminal ports exist as is the case for passenger steamers. In the case of cargo vessels, experience has led to the adoption of relatively moderate dimensions and draught of water, in order that their possible field of operations may be extended widely. Warships are designed to operate from special bases; but they must find great advantage from being able to enter and utilize other harbours or sheltered waters where the depth of water is moderate, and it is desirable also that they should be capable of proceeding to any place where an enemy can be found. Moderate draught of water is, therefore, an important feature of warship design; and the tendency in recent years to a considerable increase in the deep load draught of warships is, from this point of view, objectionable. It is notorious that some of the most recent capital ships added to war fleets cannot, when fully loaded, find shelter in harbours built primarily and at great cost to serve as naval bases, and they cannot be placed in many existing dry docks in case of injury in action because of their considerable load draughts.

XI. Enlarged dimensions enable certain advantages to be obtained outside the fundamental gains of economy in sea transport or increased speed. Amongst these advantages are maintenance of speed at sea in rough weather, greater steadiness and good behaviour in rough water, better and more spacious accommodation for a larger number of passengers, and the attraction which many passengers have toward the "biggest ships afloat." Maintenance of speed in rough seas is obviously affected by the relative dimensions of ships and waves. Ocean waves have not grown in size because larger ships have been built; and there must be an upper limit of size beyond which, so far as maintenance of speed is concerned, further increase in the dimensions of ships will have little or no effect upon regularity of service. The author is of opinion that this upper limit of size has been reached for the Transatlantic service in the *Mauretania* and her sister ship. In considering steadiness in a seaway, long experience has shown that ships of less size may be more free from heavy rolling in a seaway than much larger ships, if the latter are endowed with greater stiffness and consequently move more quickly when rolling.

XII. As larger dimensions are not essential to good behaviour at sea, and an increase in length and weight beyond the *Mauretania* not necessary to secure maintenance of speed, it would appear that the main determining factor in regard to maximum dimensions for future mercantile vessels must be found on the commercial side and not on the technical. The first cost of Transatlantic passenger steamers has been increased enormously during the last forty years. In 1874 a 15-knot steamer cost £200,000. The outlay on the *Olympic* has been put at about $1\frac{1}{2}$ million pounds sterling by men who have exact knowledge of what she has cost. The Hamburg-American and Cunard steamships now building would, on this basis, represent an expenditure of about $1\frac{3}{4}$ million pounds sterling per ship. There can be no doubt that the experienced men who guide the policy of these three steamship companies have reached the conclusion that even such huge expenditure as these figures indicate will, on the whole, prove advantageous to the owner and lead to increased revenue.

XIII. For warships, other considerations than those of first cost and cost of maintenance must determine the maximum dimensions which should be adopted. Warships are also exposed to special risks, and as a consequence it is essential to provide for considerable numbers, as well as for individual power, in the constitution of each fleet. The author's personal conviction, based upon long-continued study of the problem, is that a wiser course in warship-building would be found in a return to more moderate dimensions and a reduced unit-cost for capital ships than have recently found favour. Experience has established the fact that, without having resort to extreme dimensions, it is possible to produce capital ships which shall be powerfully armed, well protected, steady gun-platforms, capable of fighting their guns in all weathers when actions could take place, and able to maintain their speed in rough water. A reduction in the number of heavy guns carried in a single ship, as advocated by the author last year in his paper on "The Armaments of Battleships,"* would enable considerable reductions to be made in the size and cost; and for a given total expenditure on the first cost of a fleet a more formidable and flexible force could, in Sir William White's judgment, be obtained, with greatly reduced risks either from gun fire, under-water attacks, or magazine explosions.

* Page 173, No. 19, Vol. V. of *The Shipbuilder*.

SOME MILITARY PRINCIPLES WHICH BEAR ON WARSHIP DESIGN.

By Admiral Sir Reginald Custance, K.C.B., K.C.M.G., C.V.O.

(*Institution of Naval Architects ; March*, 1912.)

THE military principle advocated is the development of fire effect to the fullest extent possible. The decline in the value of armour and its possible reduction, coupled with the increased range and power of modern guns, are the changed conditions which may enable a return to be made to this old and well-tried principle.

The object of gun armament is to strike effective blows greater in number than those received. To attain this end, short-range guns were mounted in ships and forts in two, three, or even four tiers ; but this is not necessary with long-range guns, which can be spread over a wide front while the effect of their fire can be concentrated. A warship should be prepared to fight at any range. In the past this condition was satisfied by the adoption of a primary armament of long-range guns for distant firing and a secondary armament of short-range guns to increase the effect producible at "decisive ranges." With the introduction of armour this arrangement was discarded, was subsequently re-introduced, and is still a matter of controversy. We have now to decide whether it is applicable to present conditions.

It should be noted that two 12-in. guns, with mountings and ammunition, may be taken to weigh rather more than three 9·2-in. or fifteen 6-in. guns, and that the relative rapidity of hitting with the three types, under the gun-

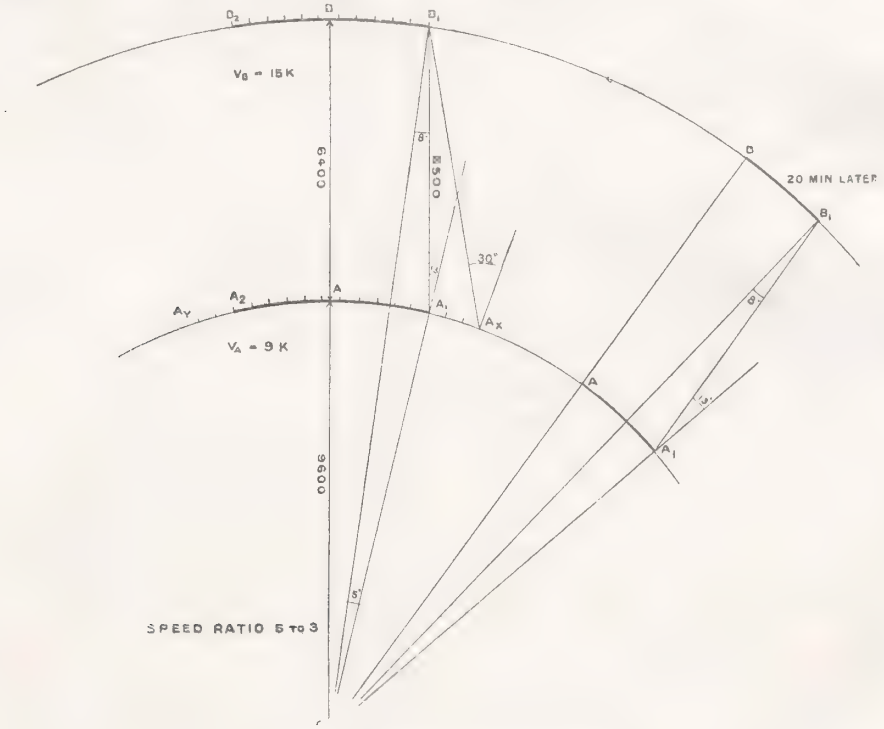


Fig. 1.—Battle of Tsu Shima.

layer's test conditions, may be taken as 1 : 2 : 3. Thus, for a given weight of armament, the relative hitting capacity would be 2 of 850 lb. : 6 of 380 lb. : 45 of 100 lb. The smallest of the three would fall off at long ranges, but after making every allowance it would seem that at decisive ranges the small guns should give a better return for their weight than the heaviest.

Decisive range may be defined as that distance at which one combatant develops in a limited time such a superiority of attack that the other is put out of action. It is a variable quantity, depending on the weather and other conditions, and has

gradually grown as guns and the appliances for using them have improved. From the "close alongside" of Nelson's time the decisive range became about 800 yards during the Crimean and American Civil Wars, about 2,500 yards at the Yalu in 1894, and inside 5,000 yards in the Russo-Japanese battles.

The determined man who desires a decision will try not only to close as quickly as possible to decisive ranges, but to maintain himself there with all guns in action at a constant range and on a constant relative bearing, which all experience has shown to be the tactical conditions required to facilitate frequent hitting. This means that the rival ships or fleets must steer in parallel courses, curved or straight, with their heads in the same direction, and, further, that broadside fire is the decisive factor.

To what extent will the curvature limit the dispersal of guns over a wide front? If a constant relative bearing and distance are maintained between the leading ships, the fleets will move on in straight lines when the speeds are equal, and in concentric circles when they are unequal. The higher the speed ratio and the shorter the range, the more curved will be the tracks. At Tsu Shima (Figs. 1 and 2), the speeds of the two fleets were 15 and 9 knots and the opening range 6,400 yards. If Rozhestvenski had brought the *Mikasa* to bear 13° abaft his beam, Togo would have been constrained to bring the *Suvorow* 8° before his beam in order to

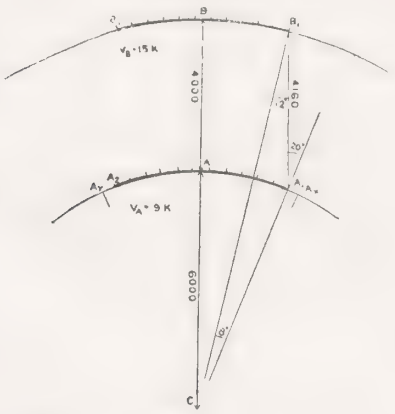


Fig. 2.—Battle of Tsu Shima.

maintain at least tactical equality. If the Japanese had steered more towards the enemy than 8° they would have closed, but dropped astern in bearing and exposed their van; if they had turned less they would have opened, gone ahead in bearing, and not fought a decisive action.

The higher the curvature, the smaller the fleets which can maintain tactical equality in this manner. As the lines close the curvature increases and throws the van and rear ships of the slow fleet out of action. At Tsu Shima, with twelve ships in each line, this would not have occurred until the range became less than 4,000 yards. But the speed ratio, 5 to 3, in that battle was very high, and it is not to be expected in future, since it means a difference of 10·6 knots if the slow fleet steams 16 knots. For very small squadrons the speed ratio might possibly reach 4 to 3, but the curvature would be immaterial in their case. For fleets of any size the ratio will perhaps be 9 to 8 and more probably unity, which means that for present-day ranges, speeds, and differences, the course may be taken as straight and the fleets unlimited in number by this consideration. Whether the speeds be equal or unequal, neither side can take position against his opponent's will on the bow of the other with all guns in action. The attempt can always be frustrated by standing on or by turning off. When two fleets once get into parallel lines, whether curved or straight, *i.e.*, into positions such as the two sides might have assumed at Tsu Shima, the only plan is to fight it out broadside to broadside. So long as there is plenty of sea room, any practicable difference in the speeds is of little importance. If land or shoals impede free movement, scope will be given for tactical skill, and the result is as likely to accrue to the advantage of the one speed as the other.

If future decisive battles are to be fought in parallel lines nearly straight, one important consequence follows. It will be possible to bring into action a large number of ships carrying possibly a greater number of guns. Fig. 3 shows this number for various ranges, when the intervals between the ships are two cables, and the gun arcs of training are 30° from the beam bearing. If the leaders remain abeam of each other and keep the range constant, A will be able to bring into action against B's rear extra ships—eight when the range is 6,000 yards, increasing or decreasing by three for each addition or reduction of 2,000 yards in the range. If the overlap is established at both ends, as might occur if B closed, the extra ships would be more numerous.

All experience points to the importance of bringing a larger number of guns into action than the enemy. In the battles between the Japanese and the Russians the former had a superiority in the number of guns of about 40 per cent. in the battle of August 10th, about 43 per cent. at Ulsan, and nearly 30 per cent. at Tsu Shima.

With regard to the accuracy of fire in battle,

the information available is necessarily imperfect. At Lissa the percentage of hits to rounds registered by the Italians was estimated at 22 per cent., by the Chilians at Point Angamos 28 per cent. In both these actions the ranges were very short. The Chinese at the Yalu seem to have made 5 per cent., and the Americans at Santiago 3·2 per cent. During the Russo-Japanese War the percentages were :—

	Russian.	Japanese.
Chemulpo...	Nil. ...	8-in guns, 10·7. 6-in. and 4·7in., 3·2.
August 10...	1. ...	6 to 7.
Ulsan	...Less than 2....	Not more than 6.

The above percentages show the need to keep the ranges not only constant but comparatively short.

Attempts to counteract the errors inseparable from firing at sea have been made in two different ways, firstly, by firing a larger number of rounds, and, secondly, by improving the accuracy of fire. The first method involves a more or less numerous battery ; while the second has led to an increase of size, a reduction in the number of the guns, and a tendency towards long ranges. In the author's opinion, this peace-tried principle of accuracy tends to undermine and destroy the war-tried principle of numbers. The latest evidence of the effect of gun fire in reducing the fighting efficiency of ships points to the necessity for discharging an increased number of rounds before a decision can be reached, especially at long ranges, and seems to indicate that the larger gun has been unduly magnified in the recent past. Again, the facts show that it is a mistake to compare the gun-power of ships by the total weights of their respective broadsides. To do so is to assume that on the average an 850 lb. 12-in. shell will damage the fighting efficiency of the ship as much as will eight 100 lb. 6-in. Such an assumption seems not to be true. The author is, therefore, of opinion that a ship of the line should be armed with two natures of guns, and that a careful balance should be maintained between the numbers and the sizes of both natures.

The use of armour for protection was first introduced about the middle of last century. Its introduction meant the partial sacrifice of armament. In the *Dreadnought* of the early seventies the armour absorbed seven times as much weight as the guns, and the number of the latter was reduced to four. Then began a reaction. In the early nineties the armour was to the guns as 3 to 1, in the year 1905 the proportion was 5 to 3 in battleships and 4 to 3 in the so-called battle cruiser. The percentage of unprotected vertical target presented by sea-keeping vessels has ranged from about 30 in the *Gloire* and 40 in the *Warrior* up to as much as 75 in some ships since built.

When ships were completely clothed with armour its military effect was to reduce the decisive range ; but with the introduction of guns of great perforating power, and of ships only

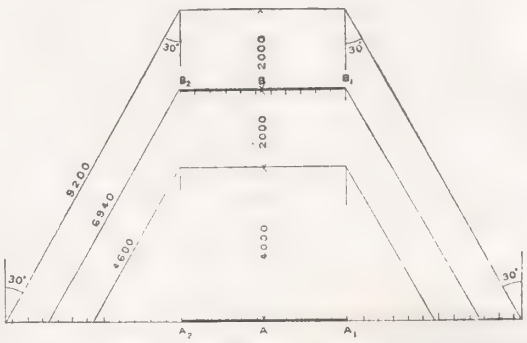


Fig. 3.

partially clothed, armour ceased to affect the decisive range, and hence ceased to influence tactics. In the Russo-Japanese and Spanish-American wars the defeated ships were generally beaten before their floating power was compromised, and the armoured belts of the victors did not exercise any decisive influence on the results. As the ship need not be made absolutely unsinkable, but only sufficiently so to win victory, should not the difficulty of hitting the waterline be carefully considered before diverting large weights from guns—the most effective form of defence—to armour, which gives imperfect protection ?

The value of thick armour as a protection for guns is not thoroughly substantiated by experience. Light armour protection for the guns against fragments of bursting shell is, however, absolutely necessary.

To show the effect of the factors of armament, armour and speed upon battleship design, the author compares two modern typical ships of the line, A and B, of about the same displacement, whose battle speeds may be taken as 17 and 22 knots respectively, and whose weights are distributed thus :—

	A Per cent. of Weight.	B Per cent. of Weight.	Per cent. Difference between B and A.
Hull	34·6 ...	37·1 ...	+2·5
Armour.....	28·0 ...	19·2 ...	−8·8
Machinery...	11·5 ...	19·3 ...	+7·8
Coal	5·0 ...	5·8 ...	+0·8
Armament...	17·3 ...	14·7 ...	−2·6
Equipment...	3·6 ...	3·9 ...	+0·3
	100·0	100·0	

SIGNOR INGRE. SALVATORE ORLANDO,
President of the Collegio degli Ingegneri Navali Italiani.

Ing. Salvatore Orlando

Both ships carry weight—in the one case of armour, in the other of armour and machinery—which gives a doubtful return in battle. Hence it may be that the changes in the size and distribution of the guns suggested and the reduction to a minimum of armour—which is all now perforable—may check the growth in the size of ships, and even reduce the present dimensions. To those reduced dimensions will correspond some most suitable speed, beyond which an increase will only be possible at the expense of fighting efficiency. The ideal fleet of ships, reduced in dimensions and cost and of corresponding suitable speed, would be more numerous than existing fleets, which is an advantage within limits, and would carry more guns and more men, which are the essentials of victory.

THE PRINCIPAL ARMAMENT AND UNDERWATER PROTECTION OF BATTLESHIPS.

By Signor Ing. Salvator Orlando.

(*Collegio degli Ingegneri Navali Italiani; November*, 1911.)

IN the first portion of his paper the author discusses the disposition of the principal armament in a battleship having eight guns in four turrets. Various arrangements which might be adopted are illustrated in Fig. 1. For each arrangement the number of guns available in any direction is indicated diagrammatically at the end of each sketch. Although broadside firing must receive the most consideration, it is important that the greatest possible number of guns should be able to fire ahead and astern. For this reason the author favours the arrangement *F*, as by it the best concentration is obtained for ahead or astern firing in conjunction with a broadside range for all the guns of 108°, a range which is not greatly inferior to the maximum which can be obtained with any of the arrangements shown.

In the second part of the paper the author deals with the problem of watertight subdivision in relation to defence against underwater attack. He strongly recommends the arrangement of protective deck and longitudinal bulkheads shown in Fig. 2, instead of the usual arrangement shown in Fig. 3. In order to compare the two arrangements, the author takes the case of a battleship of the following dimensions :—

Length between perpendiculars	512ft. 0in.
Length at waterline	542ft. 0in.
Breadth over armour	91ft. 0in.
Draught	26ft. 0in.
Metacentric height, about	6ft. 6in.

For such a vessel the inclination due to flooding one, two, and three wing compartments, having in the last case a total length of 65ft. open to the sea, is shown for the proposed arrangement in Fig. 2, and for the ordinary arrangement in Fig. 3. It will be seen that, by adopting the author's plan, the inclination is considerably reduced, *i.e.*, from 10° to 7° with three wing compartments damaged. The inner longitudinal bulkhead should be at least 16ft. from the ship's side in order to provide a reasonable chance of its not being perforated in the event of an underwater explosion at the ship's side. Should, however, a central compartment of the same length as the three wing compartments be flooded in addition to the latter, the effect upon the inclination is not great, as will be seen from Figs. 4 and 5. The extra inclining effect can be reduced to a minimum in the author's plan by keeping the flat central portion of the protective deck as narrow as possible. The author's recommendations with regard to watertight subdivision are summed up as follow :—

1.—The adoption of an arrangement of protective deck and longitudinal bulkheads, such as shown in Fig. 2.

2.—No communication doors should be fitted in the watertight bulkheads, access from one compartment to another being by the usual ladderways or by electric lifts.

3.—The central longitudinal watertight bulkhead should be abolished.

4.—The fore and aft passages for ammunition or other services should be done away with.

5.—The longitudinal wing bulkheads should be at least 16ft. from the ship's sides.

6.—The capacity of any central compartment should not exceed 1,000 tons.

7.—As no advantage would result from a triple bottom unless it were 16ft. above the outer shell, and as this depth is impracticable owing to the height to which the engines and boilers would be raised, it is better to retain a double bottom of the normal depth of about 3ft. 6in.

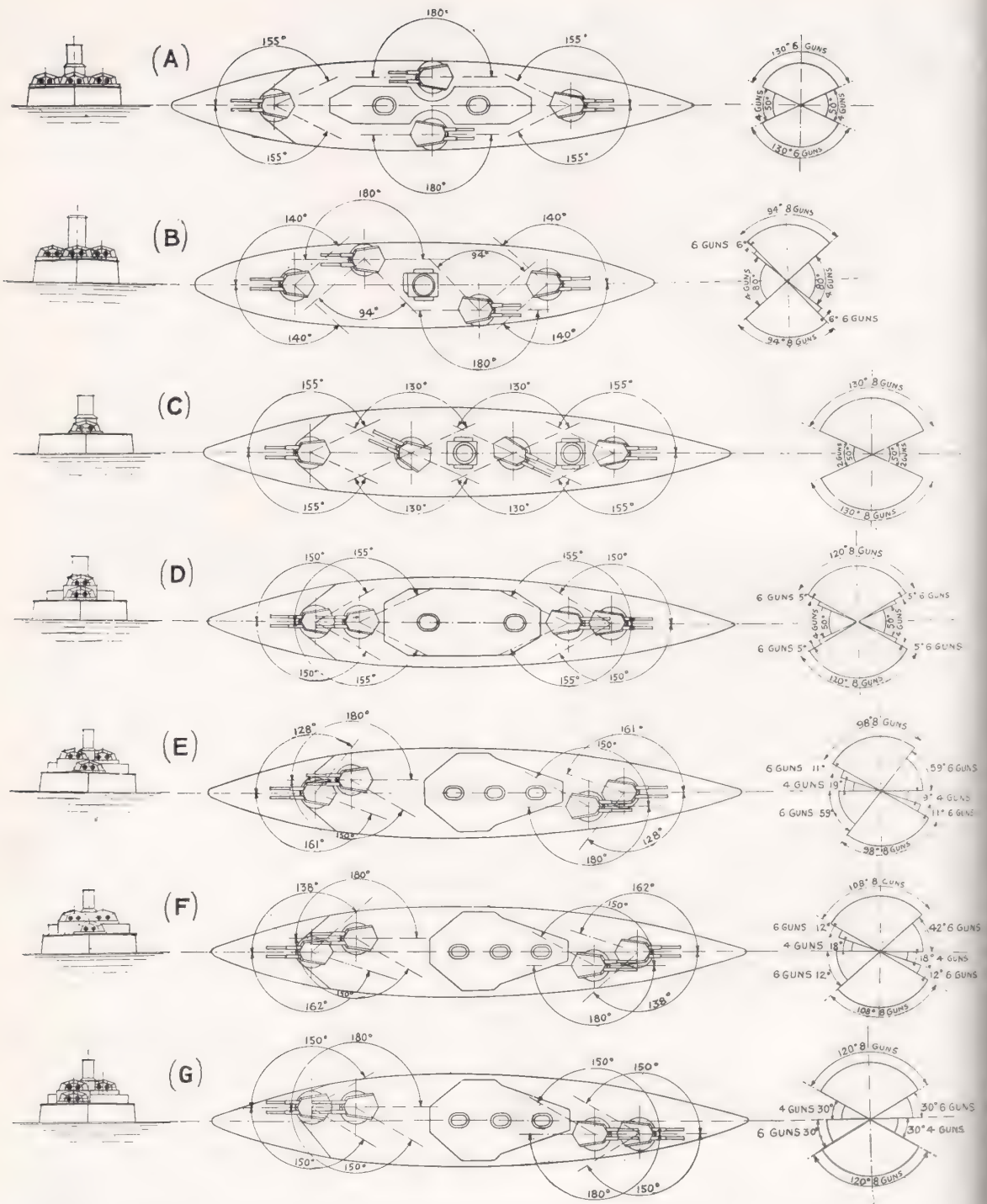


Fig. 1.—Various Arrangements of Guns for a Battleship having Eight Guns in Four Turrets.

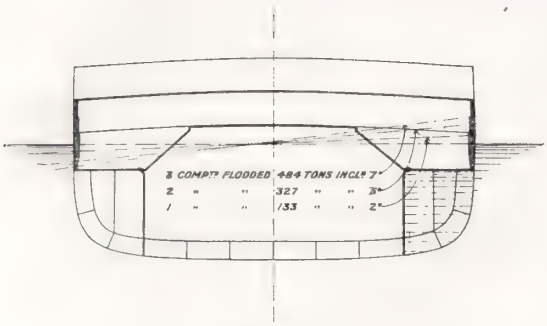


Fig. 2.

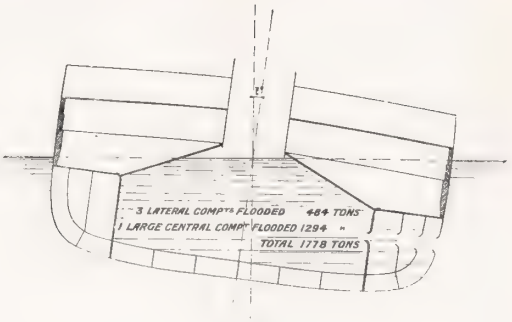


Fig. 4.

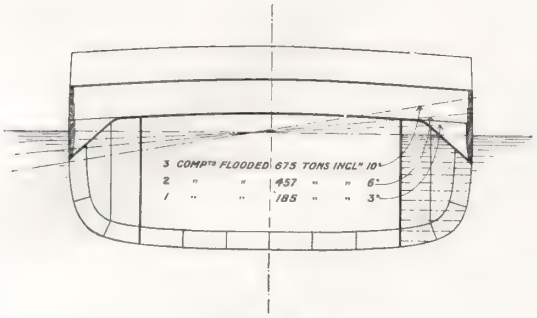


Fig. 3.

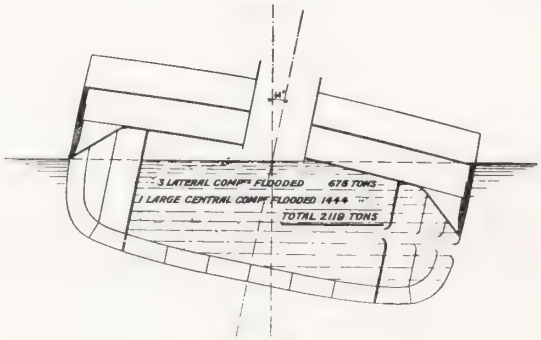


Fig. 5.

Stability and Rolling.

THE ROLLING OF SHIPS.

By Professor J. H. Biles, LL.D., D.Sc.

(*Presidential Address to the British Association, Section G; August*, 1911.)

IN his address Professor Biles reviewed the present state of knowledge with regard to the rolling of ships and the work accomplished by the Froudes, Colonel Russo, of the Italian Navy, and others in this field. The subject may be said to have been considered a solved problem, but the mysterious disappearance of several ships during recent years rendered it desirable to review the grounds of this belief, and see whether any possible combination of circumstances may cause disaster. Each ship has peculiarities of rolling due to its form, as well as to its lading, bilge keels, etc. These peculiarities, the effect they have upon rolling, and the effect different waves will have upon the rolling of the ship, can, in the author's opinion, be best studied experimentally. A machine for carrying out such experiments on lines somewhat similar to those carried out by Colonel Russo has been designed by the author, and is practically complete. In this apparatus :—

1.—Wave motion is simulated by the revolution about parallel axes of two parallel cranks of different lengths. The line joining the ends of the arms of the cranks is always in the line of the normal to the wave surface, and a line perpendicular to it is, therefore, parallel to the wave surface.

2.—From the form of the ship are determined curves, which are the shape of rollers which roll on a straight line parallel to the wave surface. The form of these rollers is such that the model of the ship in rolling maintains the position in relation to the wave surface (a) which cuts off constant volume of displacement at any angle of inclination ; and (b) in which the perpendicular to the straight line parallel to the wave surface through the point of contact is the line of the resultant of the water pressures acting on the vessel.

3.—The resistance to rolling is obtained by (a) electro-magnets, the current to which is generated by the motion of the model, and (b) a secondary electro-magnet, the current for which is in the first magnet ; (a) represents the resistance due to the angular velocity, and (b) represents the square of that velocity.

The variations in the lengths of the cranks and the speed of revolution give the variation in the wave form assumed. The variation in the electric current by resistances in the circuit gives the variation in the resistances to the rolling of the ship. For instance, the current necessary to represent the resistance of a ship with bilge keels is very different from that of one without.

THE EFFECT OF BILGE KEELS ON THE ROLLING OF LIGHTSHIPS.

Part I., by Mr. G. Idle.

(*Institution of Naval Architects ; March*, 1912.)

A LARGE quantity of data relating to the behaviour of Irish lightships during the last seven years reveals, according to Mr. Idle, the following interesting facts and phenomena with regard to maximum amplitudes of roll :—

1. The greatest rolling amplitudes are attained by the old ships of the wooden or composite class provided with double bilge logs and having generally small initial stability.

2. The largest amplitudes are reached when the sea is " breaking " (therefore steep) or, as it is termed, " confused," and the waves are advancing on the bows or quarters, and not, as might be expected, when the vessel is " beam to " a heavy swell.

3. The greatest angle of heel in a bad sea is always to the lee side, that is away from the advancing wave, no matter what the direction and force of the wind may be.

Photo by] PROFESSOR J. H. BILES, LL.D., D.Sc., &c. *[Elliott & Fry, London.*

Yours faithfully
J. H. Biles

4. Where the bilge keels are efficient it has been found that the ship's normal period of oscillation is increased by from 1 to 3 seconds, and sometimes more.

To further investigate the action of bilge keels, a series of model experiments were conducted at the William Froude National Tank on behalf of the Commissioners of Irish Lights. The model tested had a length of 12½ft., and represented, on a scale of one-eighth full size, an actual ship of the following dimensions :—

Length between perpendiculars on waterline	100ft.
Beam moulded	24ft.
Depth from top of floors to top of beam	12ft. 3½in.
Mean draught of water	9ft. 4½in.
Displacement	332 tons.
G.M.	2·37ft.
Bilge keels V-shaped (on the actual ship)	2ft. 9in. deep, 59ft. 6in. long.

The positions selected for the various bilge keels and their sectional dimensions and appearance are shown in Fig. 1. No attempt was made to imitate actual sea conditions, the relative effect upon rolling of modifications in the bilge keels being gauged by the extinction curves obtained when the model was rolled in still water.

The experiments dealt with :—

1. The absolute effect of bilge keels. The model was first rolled without keels of any sort, then with a main bar keel fitted equal to 9in. deep, and then with bilge keels added in the *C* position, equal to 2ft. 6in. deep in addition to the bar keel. Starting from a common angle of heel, the decrement of roll was found to be in the proportion of 1 : 2 : 4½ in the three cases. Tests made with the *C* keels attached, but with and without the main bar keel, gave identical results, the main keel effect being apparently eliminated when the bilge keels were in action.

2. The effect of changes of position of the bilge keels. The decrements obtained from the various positions *A*, *B*, *C*, *D*, and *E* gave, at the end of the third oscillation, the following approximate efficiency ratios :—

A keel	13·5
B „	11·5
C „	7·75
D „	6·67
E „	5·7
Bar keel alone	4·0

From these figures it will be seen that there is a sudden accession of efficiency ratio between *C* and *B*.

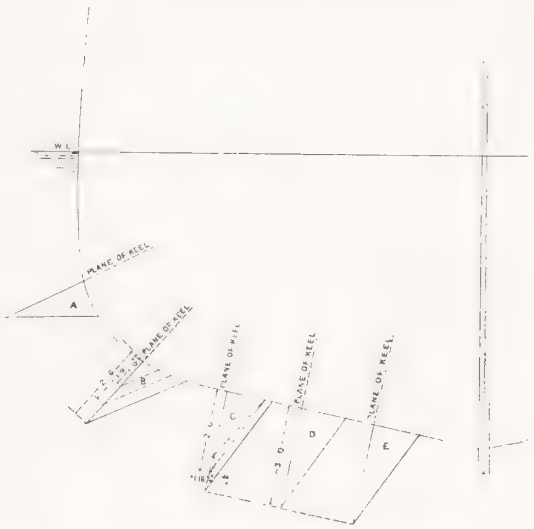


Fig. 1.—Positions and Sections of Bilge Keels.

3. The effect of variation in the depths of bilge keels. The *B* keels were reduced in depth from 2ft. 6in. to 1ft. 9in. and 1ft. respectively. The results showed that the resistance of the keel in this position increased directly as the depth.

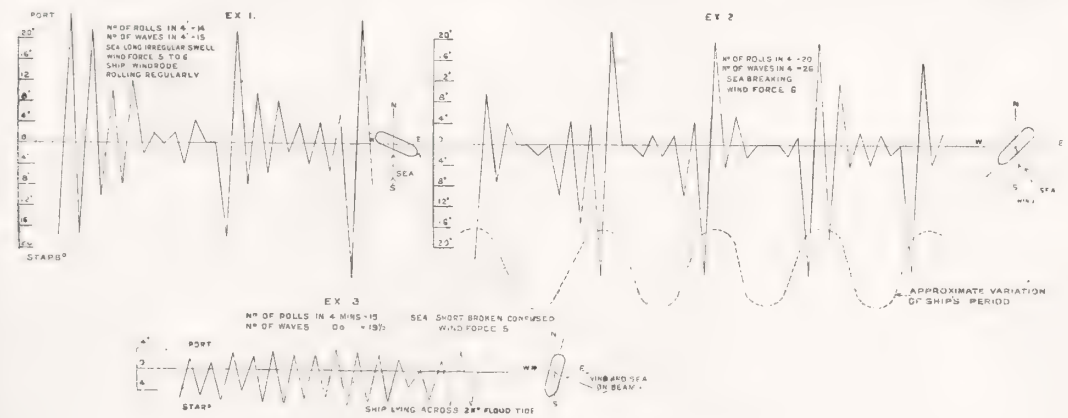


Fig. 2.—Rolling Records of Actual Lightship.

4. The effect of variation in the metacentric height. Values of the G.M. equal to 1·12ft., 2·37ft., and 3·91ft. for the ship were tried in conjunction with the bilge keels in the *B* and *C* positions. The experiments showed that the extinction of oscillations is not greatly affected even by extreme variation of the G.M.

5. The effect of adding flanges to the edge of the bilge keels, and of perforating them so as to allow the passage of water through the keels and flanges. Flanges equal to a total width of 16in. for the ship were added to the edge of the bilge keels in the *C* position, and were found to slightly diminish the extinctive effect of the keels.

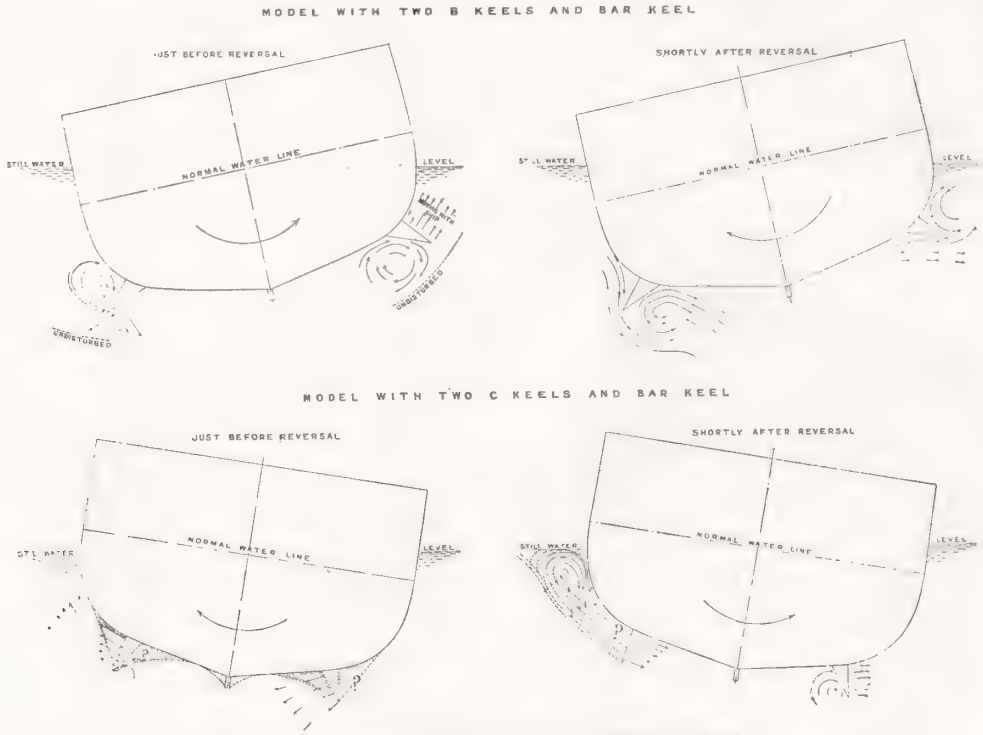


Fig. 3.—Motion of Water around Bilge Keels.

Drilling holes in the *C* keels and flanges did not alter the last-named results.

The results generally showed the inexpediency of placing bilge keels below the turn of the bilge, and if such are to be fitted a bar keel may be omitted.

Some diagrams illustrating recent rolling records taken from the actual ship (which has a G.M. of 2·65ft. and a still-water period of about 8 seconds for a double roll) are given in Fig. 2.

Part II., by Mr. G. S. Baker.

The experiments were extended beyond those required by the Commissioners of Irish Lights, in order to obtain a little more information on the action of bilge keels. From an analysis of the periods of roll recorded with and without bilge keels attached, the author estimates that the ship with 2ft. 6in. *C* keels drags with it a mass of water which, over a complete period, averages ·067 times that of the ship, more than that dragged when no keels are fitted. The rough sketches of Fig. 3 were drawn from observations, made through the observation window in the tank wall, of the rolling, when condensed milk in small quantities was dropped in to render the motion visible. The shaded patches show the solid mass of water, which, having a length equal to that of the bilge keels, would have a mass equal to ·067 times that of the ship. This dead-water carried by the keels, particularly on the faces square to the ship's side, was one of the most singular features of the experiments. Some of the water between the two *C* keels remained locked, as it were, as the ship rolled. The water moving in front of the keels was thrown off at the end of the roll. That which was thrown upwards struck the surface and returned, in the case of the *B* keel, to form the large eddy at the back of the keel, and, in the case of the *C* keel, to strike the

ship's side and glide off at the turn of the bilge, never, however, actually reaching the keel again.

It is generally held that the decrement of roll per single swing of the ship is given by the equation

$$-\delta\,\theta = a\,\theta + b\,\theta^2$$

θ being the mean angle of swing port and starboard, and a and b coefficients, which depend, the former on the wave-making resistance, and the latter on friction and head resistance. A considerable number of the curves of declining angles were differentiated to obtain the a and b values, which are given in Table I.

It was found that the formula

$$-\delta\,\theta = a\,\theta + b\,\theta^2$$

when the model was fitted with bilge keels held for rolls of quite moderate amplitude only. The curve of $\delta\,\theta$ begins to fall off in almost every case at the large amplitudes. The angles at which this begins are approximately :—10° (from the upright) in the *C* position, 8° in the *B* position, and 6° in the *A* position. When the model was naked, or fitted with the bar keel only, the law held very closely for amplitudes up to 15°.

From considerations explained in the paper, the author suggests the inclusion of a θ^3 term in

Table I.

Position of Keels.	Depth of Keels.	Period T. Secs.	G. M.	T^2m.	Initial.		Small.		Larger roll.	
					*a.**	*b.**	*a.*	*b.*	*a.*	*b.*
	Ft. In.		Feet.							
A	2 6	4·29	2·37	43·6	·14	·024	·01 to 0	·09 to ·05	negative	·08
B	2 6	4·17	2·37	41·3	·09	·022	·015	·043	0	·047
C	2 6	4·30	2·37	43·8	·04	·01	·01	·0193	·01	·0193
D	3 0	4·23	2·37	42·4	·065	·013	·064	·013	·10	·01
E	3 0	4·22	2·37	42·2	·04	·01	·058	·009	·058	·009
B	2 6	4·17	2·37	41 24	·09	·022	·015	·043	0	·047
B	1 9	4·24	2·37	42·6	·05	·018	·036	·022	·036	·022
B	1 0	4·27	2·37	43·2	0	·022	·07	·01	·07	·01
B	2 6	3·39	3·91	44 9	·09	·02	·02	·035	0	·037
B	2 6	4·17	2·37	41·2	·09	·022	·015	·043	0	·047
B	2 6	6·36	1·12	45·3	·095	·024	·013	·042	·013	·042
C	2 6	3·35	3·91	43·9	·035	·009	·036	·014	·02	·015
C	2 6	4·30	2·37	43·8	·04	·01	·012	·0193	·012	·0193
None fitted	—	4·016	1·54	24·84	—	—	·025	·004	·025	·004
As above with linen cloth for 5′ 6″ of length.		4·02	1·54	24·83	—	—	·025	·0064	—	—
Bar only	0 9	4·29	1·54	28·3	·025	·011	·029	·012	·046	·011
C	2 6	4·20	1·54	27·2	·07	·028	·031	·037	·031	·037

* For θ in degrees and $\delta\,\theta$ for roll from out to out.

The "*a*" "*b*" values under the heading "Initial" are obtained by producing the curves to zero angle and are put forward only as rough approximations.

the above formula, and the curves for the 2ft. 6in. keels at different immersions treated on these lines give the following results :—

TABLE II.

Position of Keels.	a.	b.	c.
A	·07	·04	·003
B	·063	·0307	·00085
C	·012	·0193	0

The relative importance of the two positions of keels *B* and *C* when the ship is rolling amongst waves is shown in Fig. 4. The curves of amplitude have been obtained by graphical integration, the ship being assumed upright and at rest momentarily in the trough of a wave having a period of 4·75 seconds and a maximum slope of 10°, and the ship's period being taken at 4·25 seconds. It can be seen from the diagram that the ship's amplitude is 33 per cent. less with keels in the *B* position than it is with the same keels in the *C* position.

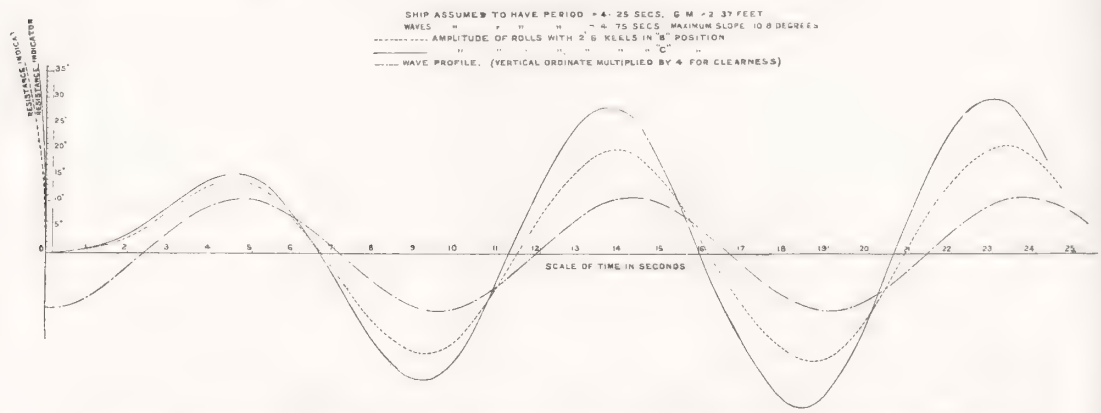

SHIP ASSUMED TO HAVE PERIOD = 4·25 SECS. G M = 2·37 FEET
WAVES " " " = 4·75 SECS. MAXIMUM SLOPE 10·8 DEGREES
AMPLITUDE OF ROLLS WITH 2' 6" KEELS IN "B" POSITION
" " " " " "C" "
WAVE PROFILE. (VERTICAL ORDINATE MULTIPLIED BY 4 FOR CLEARNESS)
RESISTANCE INDICATOR
SCALE OF TIME IN SECONDS

Fig. 4.—Relative Effect of Bilge Keels on Rolling of Lightship amongst Waves.

RESULTS OF CALCULATIONS REGARDING THE EFFECT OF AN INTERNAL FREE FLUID UPON THE INITIAL STABILITY AND THE STABILITY AT LARGE ANGLES IN SHIPS OF VARIOUS FORMS.

By Mr. A. CANNON.

(*Institution of Naval Architects ; March*, 1912.)

THE paper presents the results of some preliminary investigations made by the author in connection with research work upon which he is engaged as holder of the Post Graduate Scholarship in Naval Architecture awarded by the Commissioners for the International Exhibition of 1851, relating to the rolling of ships amongst waves of varying periods, and the effect of loose water in double bottom tanks, anti-rolling chambers, etc. In all cases the density of the internal fluid is taken to be the same as the external fluid.

Considering first the effect of free water upon initial stability, the author points out that the problem can be attacked in two ways, (1) by considering the free water as an added weight, and (2) by considering the water as a loss of buoyancy. Different values for the virtual decrease in metacentric height are obtained according to the method followed. The change in metacentric height in the case of the first method is for a volume $V + v$, including the free water ; while in the case of the second method, the change in metacentric height is for a volume V, excluding the free water. The decrease in the stability moment at a small angle is the same by either method, and must necessarily be so.

In general practice, if the amount of water that enters is fixed and does not vary with the angle of heel of the ship, the added weight method is

the better one. Where the amount of water varies with the heel, it is better to adopt the second method and consider it as a loss of buoyancy. In the cases of initial stability considered in the paper, the loss of buoyancy method has been adopted, since it lends itself more readily to the question of considering the effect produced by varying the depth of the internal water.

Attention was first directed to bodies of simple regular shapes, as follows :—

Case 1. A rectangular box-shaped vessel 100ft. long, by 50ft. broad, by 20ft. draught, with no

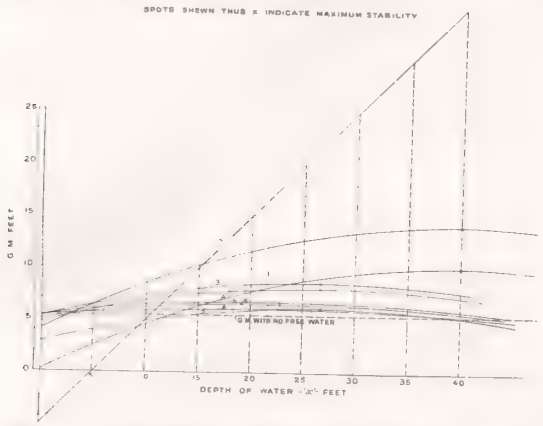


Fig. 1.—Curves of Initial Stability for Rectangular Vessel containing Free Water.

free water, with its centre of gravity 15ft. above the keel, and containing a rectangular-shaped centre line compartment into which the water is steadily poured. Curves showing the value of G.M. in terms of depth of water in the compartment are shown in Fig. 1, and correspond to the following conditions :—

In the case of Nos. 3, 6 and 9 there is practically from the first a gain in the stability, showing that the effect of free water in a narrow compartment

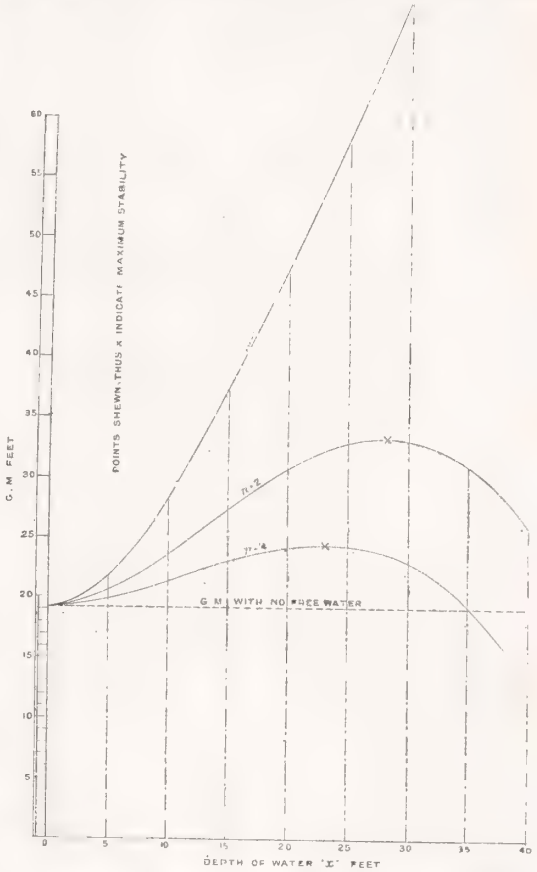


Fig. 2.—Curves of Initial Stability for Triangular Vessel containing Free Water.

No. of Curve.	Length of Compartment containing free water.	Breadth of Compartment.
1.	Total length of vessel	Total breadth of vessel
2.	,, ,, ,, ...	Half ,, ,,
3.	,, ,, ,, ...	One-quarter breadth of vessel
4.	Half ,, ,, ...	Total ,, ,,
5.	,, ,, ,, ...	Half ,, ,,
6.	,, ,, ,, ...	One-quarter ,, ,,
7.	One-quarter length of vessel	Total ,, ,,
8.	,, ,, ,, ...	Half ,, ,,
9.	,, ,, ,, ...	One-quarter ,, ,,

in the bottom of the vessel as regards initial stability is a help rather than the reverse.

Case 2. A vessel of triangular section, floating with vertex downwards, and having a breadth at the waterline of 50ft., draught of 20ft., and C.G. above keel 15ft. The results are shown in Fig. 2. The length of the compartment is taken $\frac{1}{n}$ the length of the ship and the cases considered are those in which $n = 1$, $n = 2$, and $n = 4$.

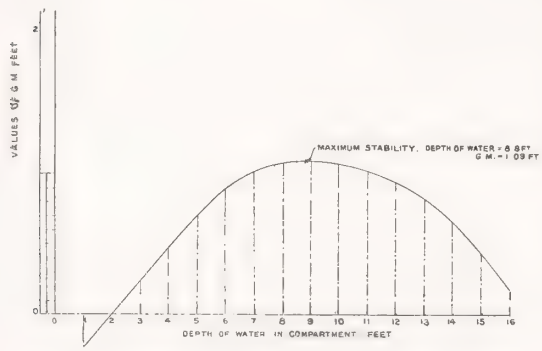


Fig. 3.—Curve of Initial Stability for Cargo Vessel having a long Compartment into which Free Water is slowly admitted.

For practicable depths there is always a gain in stability, and it is upon this principle that lifeboats are often built with triangular shaped sections.

Case 3. A vessel of parabolic section, floating with axis vertical and vertex downwards, of the same dimensions as in Case 2. The results obtained were similar in character to Case 2.

It was found that in all three shapes considered the maximum values of the metacentric height occur when the internal water is at the same level as the external water.

Having studied these simple cases a ship-shaped form of the following dimensions was considered :—

Length between perpendiculars	182ft.
Breadth moulded	27ft.
Depth moulded to main deck	13ft. 5in.
K.G.	9·8ft.
G.M.	3·0ft.
Draught	5ft. 9in.
Displacement	600 tons.

A compartment 50ft. long and extending from the outer bottom to the main deck is assumed to be empty and water then slowly admitted. Fig. 3 shows the G.M. plotted against depth of internal water, and, as in the simple shapes, the initial stability is a maximum when the level of the water inside and outside is the same.

For practical purposes, however, the best criterion of stability is the $G Z$ curve, as it is of very little use to have a large G.M. with a small statical stability curve. A number of curves of $G Z$ were drawn as obtained for various depths of the internal water. The values were plotted for a total displacement, including the internal water, of $W + w$, it being more convenient in this case to treat the internal water as weight added than as buoyancy lost, and in all cases showed a very large reduction in the stability, due to freedom of the surface.

In order to examine and analyse the results, the cross curves of $G Z$ were prepared as shown in Fig. 4. These curves are different from the ordinary cross curves, in that the position of G is not constant, being only constant for the same depth of water. From this figure it will be seen

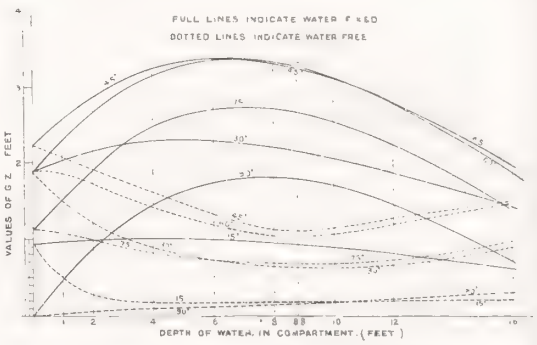


Fig. 4.- Cross Curves of G Z.

that the best depth for initial stability, *viz.*, 8·8ft., is the worst depth for stability at large angles, and also that at all depths less than this the stability is the greater the lesser the depth.

Thus, in dealing with the statical effect of free water in a ship, although for small quantities of water the initial stability is very greatly reduced —in some cases even to a negative value—yet the statical stability at large angles is the better the smaller the quantity of free water.

THE SAFETY OF SHIPS AT SEA.

By Professor W. S. Abell.

(*Liverpool Engineering Society ; December*, 1911.)

THE safety of ships at sea must be influenced by two considerations :—(*a*) Whether the control of the vessel is injured in any way owing to the breakdown of the engines, steering gear, or rudder ; or (*b*) whether the buoyancy is impaired by flooding caused by loss of hatches, by working of the structure, or by collision. In case of damage by collision only the highest class of ships with efficient watertight sub-divisions are likely to survive anything but moderate weather.

In the general case of a ship at sea in bad weather, it is always possible, so long as control is maintained, to steer such a course that the movement of the ship is considerably relieved, and *in extremis* the vessel could be hove to with her head to the wind and sea. If control is no longer possible, the vessel takes up a broadside position to the waves and then considerable rolling must result. The safety of the ship in such a condition depends on the energy required to upset her. In other words, she will capsize if the work impressed on her by external forces, *viz.*, the wind and the motion of the waves, is greater than the energy which she possesses to resist such action.

In the case of the motion of the ship among waves and in particular broadside thereto, the stability as calculated for the ordinary still-water condition is reduced. The effect of wave motion is to diminish the weight of the vessel when on the crest of a wave and to increase the weight when in the hollow. A vessel struck by a squall therefore offers less resistance on the top of a wave than in still water. Another point is that the resultant forces acting on a vessel rolling among waves are not vertical, but parallel to the normal to the wave surface. Consequently, for a given inclination of ship to the vertical, the restoring couple is less than in still water. When the period of recurrence of the wave and the period of oscillation of the ship very nearly coincide, the wave transmits energy to the ship at every roll ; this energy must accumulate in the vessel and ultimately she must capsize. Such a contingency is, however, remote, for except in a long swell the periods of successive waves vary considerably.

The stability of a vessel involves two quantities, that due to form which must depend on the proportions of the vessel, and that due to weight which depends on the height of the centre of gravity. The stability due to form is fixed when dimensions are given, while the stability due to weight can be varied by altering the disposition of the cargo. It is therefore necessary either to design the vessel so that the centre of gravity will not be too high, or, where the dimensions are restricted, to arrange rules for loading to achieve the same object. The stability due to form is increased both as regards range and area when either beam or freeboard is increased, the depth of watertight structure or moulded depth being constant. Now the load freeboard for large and medium sized vessels is a constant fraction of the depth, and consequently cannot be altered. The only manner in which stability can be improved is by increasing the ratio of beam to depth. If, therefore, the ship is to be safe in all conditions, it is necessary in the design to estimate the probable highest position of the centre of gravity of the ship, allowing for deck erections and cargo carried therein or on deck. The proportion of beam to depth should then be fixed in order to ensure sufficient stability.

As regards the stability due to weight, this depends solely on the vertical position of the centre of gravity, which in turn depends on the mean depth of the structure, omitting for the time consideration of superstructure above the upper or weather deck. The depth of structure fixes the centre of gravity of the vessel in the light condition, and also, generally speaking, the centre of gravity of the cargo. The addition of high central superstructure to a flush-deck vessel with moderate erections considerably raises the centre of gravity of the vessel, both light and loaded, and the same result follows if cargo be carried on deck. Central citadels have another disadvantage, that greater surface is offered to wind pressure and that the point of application of this upsetting force is raised.

It would appear in such cases that, as compared with the flush-deck type of ship, it is necessary to increase the ratio of breadth to depth as measured in the ordinary way, or, on the other hand, to keep the same ratio of the two quantities, and increase the value to be taken for the depth by adding to it a factor representing the deck erections. The addition to the height involved and the increased amount of deck erections, as far as stability is increased, brings this type of vessel somewhat back to that of the narrow and

deep three-deck vessel, which was largely built between 1870 and 1880. A consideration of the experience obtained with this type indicates that for flush-deck ships the proportion of breadth to depth of 1·45 is unsafe, and that a ratio of 1·6 appears to give sufficient stability. It may perhaps be safe to make a somewhat general statement, and say that the case of a flush-deck vessel, where the beam is not more than 1·5 times the depth to weather deck, requires investigation as regards its general safety. For vessels with central superstructures, the beam must, of course, bear a greater ratio.

The value of the maximum righting couple and the range of stability required to ensure safety are matters of opinion and depend upon a knowledge of the condition of similar ships which have

vessel is inclined by a sudden squall of the same intensity of pressure, she will heel over beyond this angle until her energy of motion is overcome, *i.e.*, until the area θAH = area ABk, and if the wind pressure be sufficient the work done by the squall may be sufficient to capsize the ship. The conditions are more unfavourable when the vessel is rolling in a seaway. The worst case to take is that in which a squall strikes the ship at the moment she is situated on the crest of a wave and has just completed a roll to windward.

Assuming the side area of the vessel to be calculated in the manner already stated, the author has worked out, as an example, the intensity of wind pressure necessary to capsize the ship, of which the stability curves are given in Fig. 1, both in the load and light conditions, at rest and at the

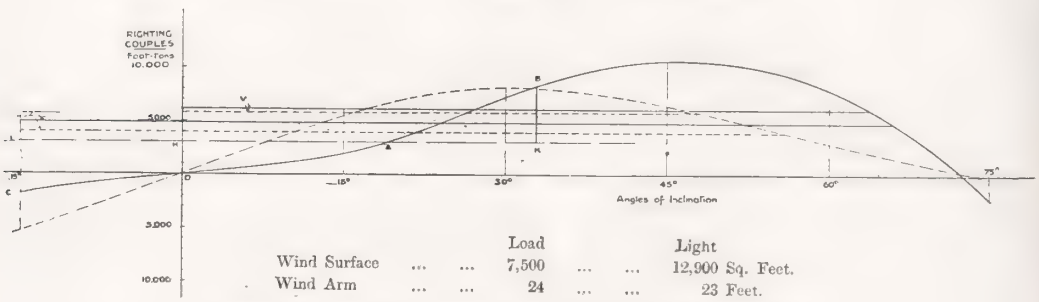


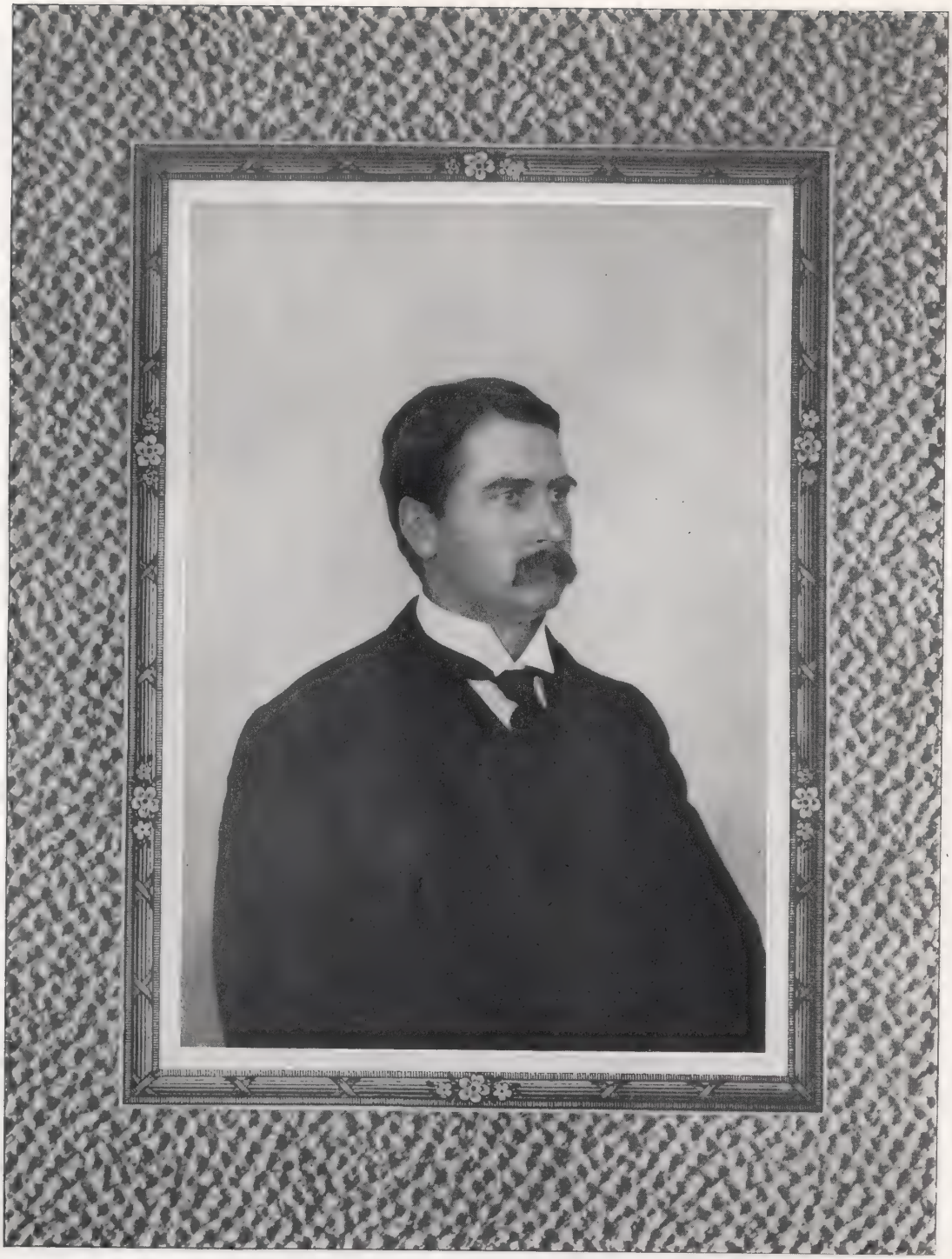
Fig. 1.—Effect of Wind Pressure upon Stability.

survived very bad weather. One of the most important considerations in connection with this problem is the extent and height of the vessel above water exposed to the wind. It is difficult to estimate the exact value of the wind pressure to be expected on a structure such as a ship. For relative calculations, in comparison with ships of which the stability and condition at sea are fairly well known, the area supposed to be acted upon by the wind can, however, be taken to include only the main features of the vessel, such as the side above water, the various deckhouses, and the funnels.

If a vessel be upright in still water and the wind rise gradually, the ship inclines to a steady angle A (Fig. 1) such that the righting couple possessed by the ship balances the upsetting moment of the wind, OH. If, however, the

completion of a roll of 15° to windward. It was found that for the condition at rest the maximum and minumum capsizing pressures are 75 and 44lb. per square foot respectively. With the ship at the end of a roll of 15° to windward, and just poised on the crest of a wave, the wind pressure necessary to capsize becomes 48lb. per square foot when fully loaded, and only 24lb. per square foot when in the light condition.

Supposing an extra tier of deckhouses has been added on the vessel, the gust pressure necessary to capsize the ship in still water is reduced from 75 to 60lb. in the load condition, and from 44lb. to 36lb. in the light condition. It would therefore appear that in this vessel no additional superstructure could be carried without an increase in beam.



NAVAL CONSTRUCTOR D. W. TAYLOR, U.S.N.

D. W. Taylor
Naval Constructor - U.S.N.

Resistance and Propulsion.

SOME MODEL BASIN INVESTIGATIONS OF THE INFLUENCE OF FORM OF SHIPS UPON THEIR RESISTANCE.

By Naval Constructor D. W. Taylor, U.S.N.

(*American Society of Naval Architects and Marine Engineers; November*, 1911.)

IN this paper Mr. Taylor gives the results of some experimental investigations made at the United States model basin of the effect of shape toward the extremities upon resistance in the case of two series of models, one series having a longitudinal coefficient of ·60 and the other a longitudinal coefficient of ·64. The length of the models was 20ft., their midship section coefficient ·96, and the ratio of beam to draught 2·5 in all cases. Each series consisted of 16 models, four different curves of sectional area each being tried with four different water lines. In each case the displacements of the fore and after bodies were equal, but the waterplane had a greater area abaft the centre line, in order to approach more closely average practice. The models were made in halves, so that the bows and sterns could be interchanged. The general character of the sectional area and waterline curves tried will be seen from Fig. 1, which shows the curves for the series having ·60 longitudinal coefficient.

The resistance being partly due to surface friction and partly to the disturbance created, the first question investigated was the relative surface area of the various models. It was found that the effect of practicable variations of shape of sectional area curves and waterline fineness upon wetted surface is comparatively small, the difference between extreme cases not being more than about 2 per cent.

The total resistance curves of the two extremes and an intermediate form of each series are given in Fig. 2. Bearing in mind that comparatively few vessels are driven at speeds greatly above that corresponding to 4 knots for the 20ft. model, the author points out that the variation of resistance corresponding to very radical variations of form is after all not great. It is interesting to note the critical speed, a little above 4 knots, at which the models change place materially. Models Nos. 1092 and 1191, which are of the extreme type, combining full-ended sectional areas and fine-ended waterlines, are the worst below 4 knots, while between 4 and 5 knots they show themselves the best. This is a typical result, and shows that the features of a model should be adapted to suit the speed.

With regard to residuary resistance alone, Fig. 3 shows the residuary resistance in pounds per ton displacement plotted on speed-length ratio for a number of models of the ·60 longitudinal coefficient series grouped by curves of

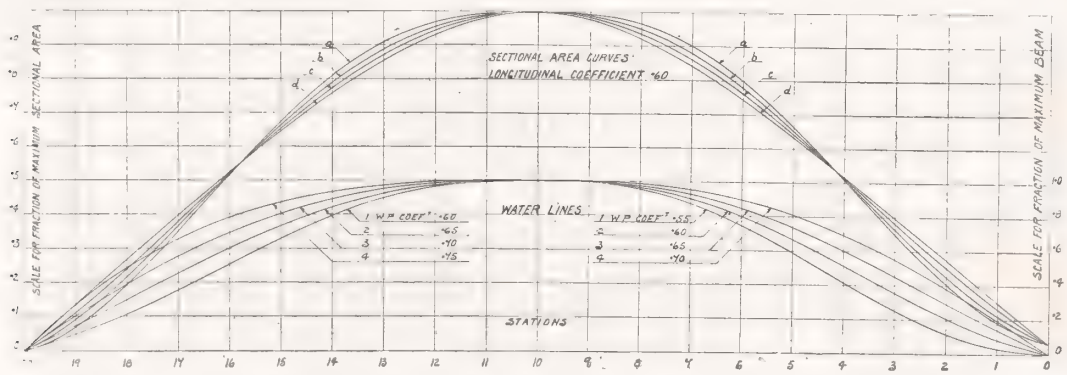


Fig. 1.—Curves of Sectional Areas and Waterlines for Series of Models with ·60 Longitudinal Coefficient.

sectional area, while Fig. 4 shows the same curves grouped by waterlines. Considering Fig. 3, in every case the fine bow waterline appears to be the best below the speed length ratio of ·9. Moreover, the fine bow waterline in combination with the full stern waterline appears better than the fine bow in combination with the fine stern. At higher speeds the fine bow with the full stern loses its superiority. Considering Fig. 4, broadly speaking the fine-ended curves of sectional area have the better of it at and below a speed length ratio of about ·9, while at the high speeds they areas aft can be considerably varied without material effect upon resistance. Grouped by curves of sectional area, for moderate speeds, the results again show comparatively small variation, the fuller waterline aft with the finer ended curve of sectional areas, leading to a V-shaped stern, being somewhat superior. At about a speed length coefficient of ·95, however, a change in the results occurs, and in every case the fine waterline aft is the best at high speeds.

With regard to the best combination of sectional area and load waterline curves, in the case of the

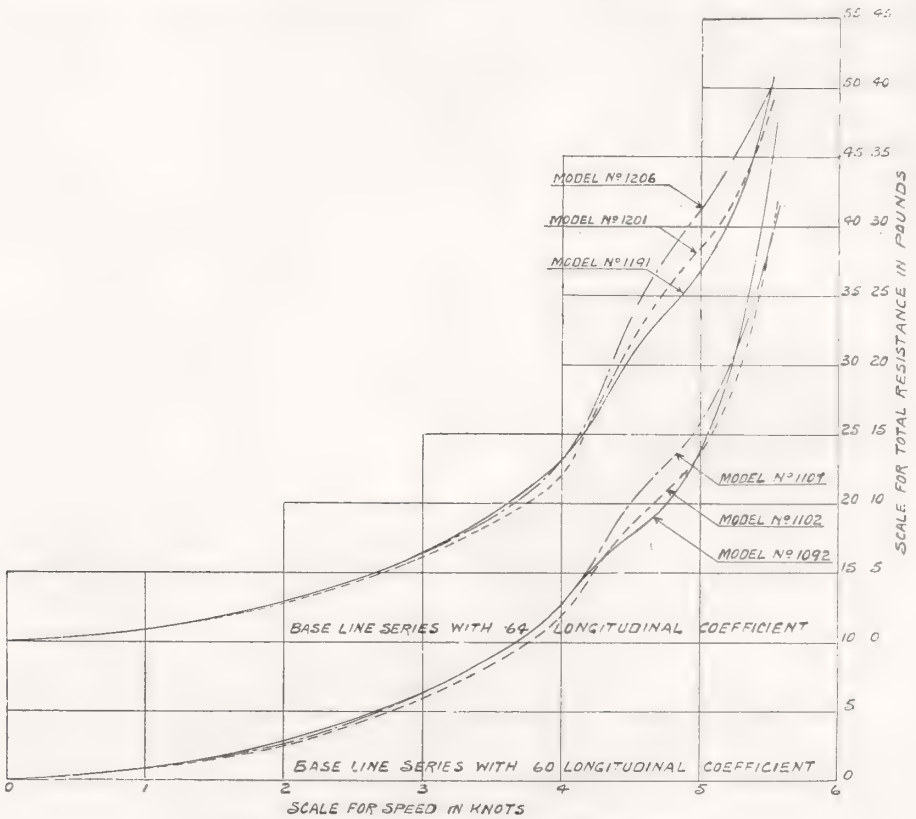


Fig. 2.—Typical Curves of Total Resistance.

are the worst in every case. Similar results, although not so marked in character, were obtained for the ·64 longitudinal coefficient series.

Undoubtedly the resistance of a given stern will be different as it is combined with varying bows, and *vice versâ*. It was not practicable to test all bows with all sterns, but some investigation was made of the stern effect by combining a single bow with all sixteen sterns of the same series. Grouped by waterlines, the results show remarkably small variation for all speeds, thus indicating that the shape of the curve of sectional

·60 longitudinal coefficient series, at and below a speed length ratio of ·9, there appears to be a distinct region for minimum residuary resistance, corresponding to quite fine ends on the curves of sectional area, and to a bow waterline coefficient slightly greater than the longitudinal coefficient in conjunction with a full after waterline. In the case of the series having ·64 longitudinal coefficient, the region of minimum residuary resistance for low speeds is not so well defined, and appears to correspond to a relatively smaller bow waterline coefficient than in the ·60 series.

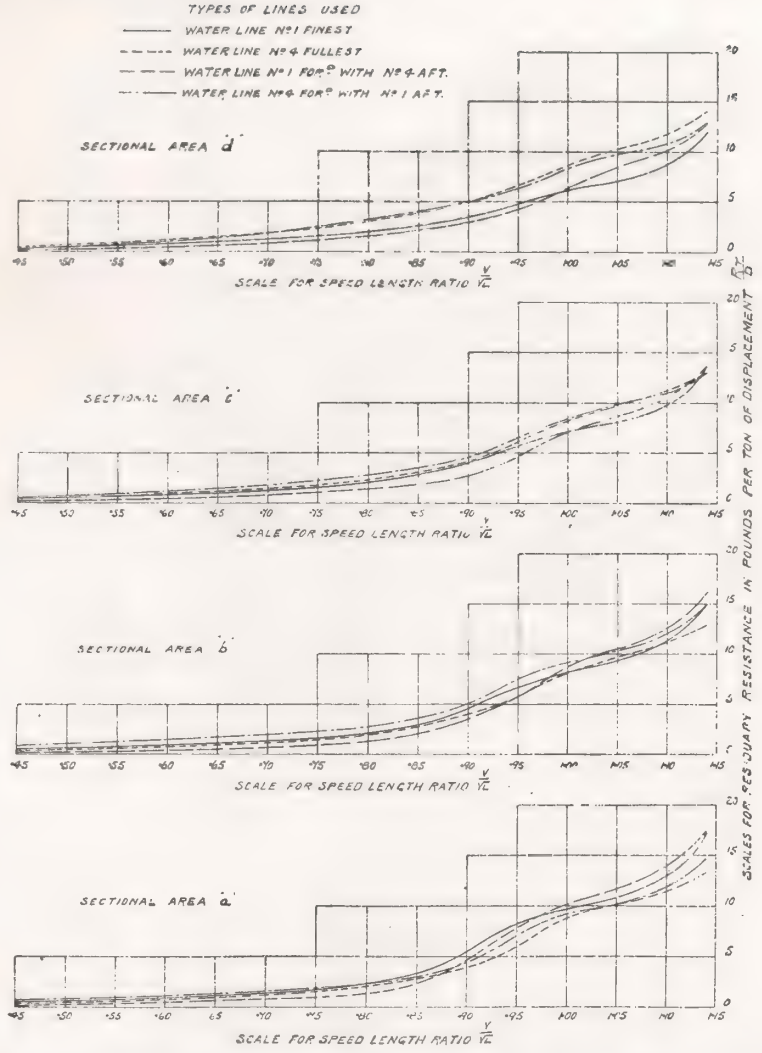

TYPES OF LINES USED
WATER LINE No 1 FINEST
WATER LINE No 4 FULLEST
WATER LINE No 1 FOR. WITH No 4 AFT.
WATER LINE No 4 FOR. WITH No 1 AFT.
SECTIONAL AREA d
SECTIONAL AREA c
SECTIONAL AREA b
SECTIONAL AREA a
SCALE FOR SPEED LENGTH RATIO V/√L
SCALES FOR RESIDUARY RESISTANCE IN POUNDS PER TON OF DISPLACEMENT

Fig. 3.—Residuary Resistance of various Combinations of Sectional Areas and Waterlines, grouped by Curves of Sectional Area.

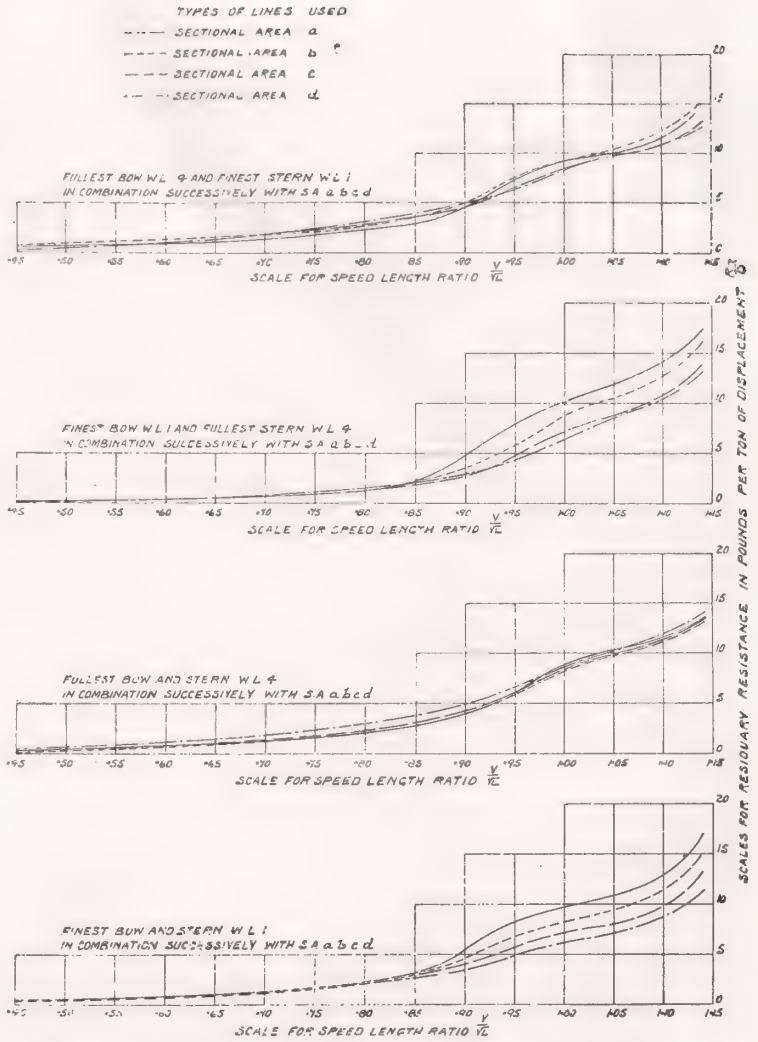

TYPES OF LINES USED
SECTIONAL AREA a
SECTIONAL AREA b
SECTIONAL AREA c
SECTIONAL AREA d
FULLEST BOW W.L 4 AND FINEST STERN W.L 1
IN COMBINATION SUCCESSIVELY WITH S.A a b c d
FINEST BOW W.L 1 AND FULLEST STERN W.L 4
IN COMBINATION SUCCESSIVELY WITH S.A a b c d
FULLEST BOW AND STERN W.L 4
IN COMBINATION SUCCESSIVELY WITH S.A a b c d
FINEST BOW AND STERN W.L 1
IN COMBINATION SUCCESSIVELY WITH S.A a b c d
SCALE FOR SPEED LENGTH RATIO V/√L
SCALES FOR RESIDUARY RESISTANCE IN POUNDS PER TON OF DISPLACEMENT

Fig. 4.—Residuary Resistance of various Combinations of Sectional Areas and Waterlines, grouped by Waterlines.

In each case for speed length ratios higher than from ·85 to ·90 there is a marked change in the results. For $\frac{V}{\sqrt{L}} = 1{\cdot}1$, for instance, the minimum residuary resistance corresponds to the fullest ended curves of sectional areas, while the corresponding bow waterline coefficient falls off materially. In other words, for the minimum resistance at high speeds the ship should be considerably fined at the waterline and filled out below water.

The quantitative effect in actual cases of the variation of form considered would not be more than some 5 per cent. in passing from the worst likely combinations of waterline and sectional area curves to the best.

The experiments show the bulbous type of bow to be distinctly superior for high speeds where $\frac{V}{\sqrt{L}}$ is from 1 to 1·2, and the author sees no reason why in many cases some approach to the bulbous type could not be made with advantage.

EXPERIMENTS ON THE "FROUDE."

By Professor C. H. Peabody.

(*American Society of Naval Architects and Marine Engineers; November*, 1911.)

THE inception of these experiments was due to the late Dr. Charles G. Weld, of Boston, who undertook to provide the means, while entrusting the execution of his ideas to the author, in conjunction with the staff of the Department of Naval Architecture and Marine Engineering of the Massachusetts Institute of Technology. The prototype chosen for investigation was the Revenue Cutter *Manning*, which was tested by the author under favourable conditions in 1899. The scale of the model was made one-fifth that of the *Manning*, the leading dimensions being :—

	Manning.	*Froude.*
Length B.P., feet	188·0	37·60
Moulded beam, feet	32·0	6·40
Mean draught, feet	12·3	2·52
Displacement, tons	1,000	8·00
Wetted surface, sq. ft.	727·3	29·60
Diameter propeller, feet	11·0	2·20
Pitch propeller, feet	12·33	2·47
Projected area of blades, sq. ft.	31·4	6·28

The general arrangement of the *Froude* is shown in Fig. 1. The vessel is indirectly propelled by a gasoline electric generating set and an electric motor with a chain drive to the propeller shaft. Both generator and motor have separate excitation on the Ward-Leonard system, which gives great facility and range of control and provides for electric measurements of power.

The experiments were made during the summer months of 1910 and 1911, and consisted in making series of progressive speed trials over a measured course one-eighth of a knot long. As an unexpectedly large influence of the wind was found even on fair summer days, the observers began work early in the morning, from four to eight o'clock being favourite hours. Runs were made with and against the wind in all cases, and the true results, corrected for wind resistance, were obtained by the method explained by Professor Everett in an appendix to the paper. A continuous recording device was used to register the propelling revolutions, thrust, time, and signals at the beginning and end of the run. The thrust of the propeller was weighed directly by means of a bent lever arrangement in which the major part of the thrust was counterbalanced by a large sliding weight, but to allow for and measure minor fluctuations the extreme end of the bent lever bore against an indicator spring which was secured to the frame of the boat.

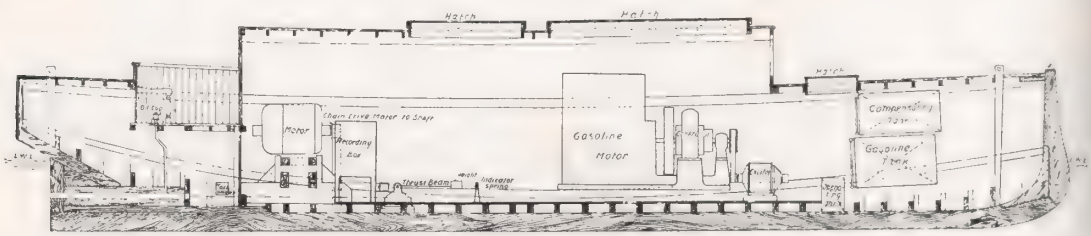
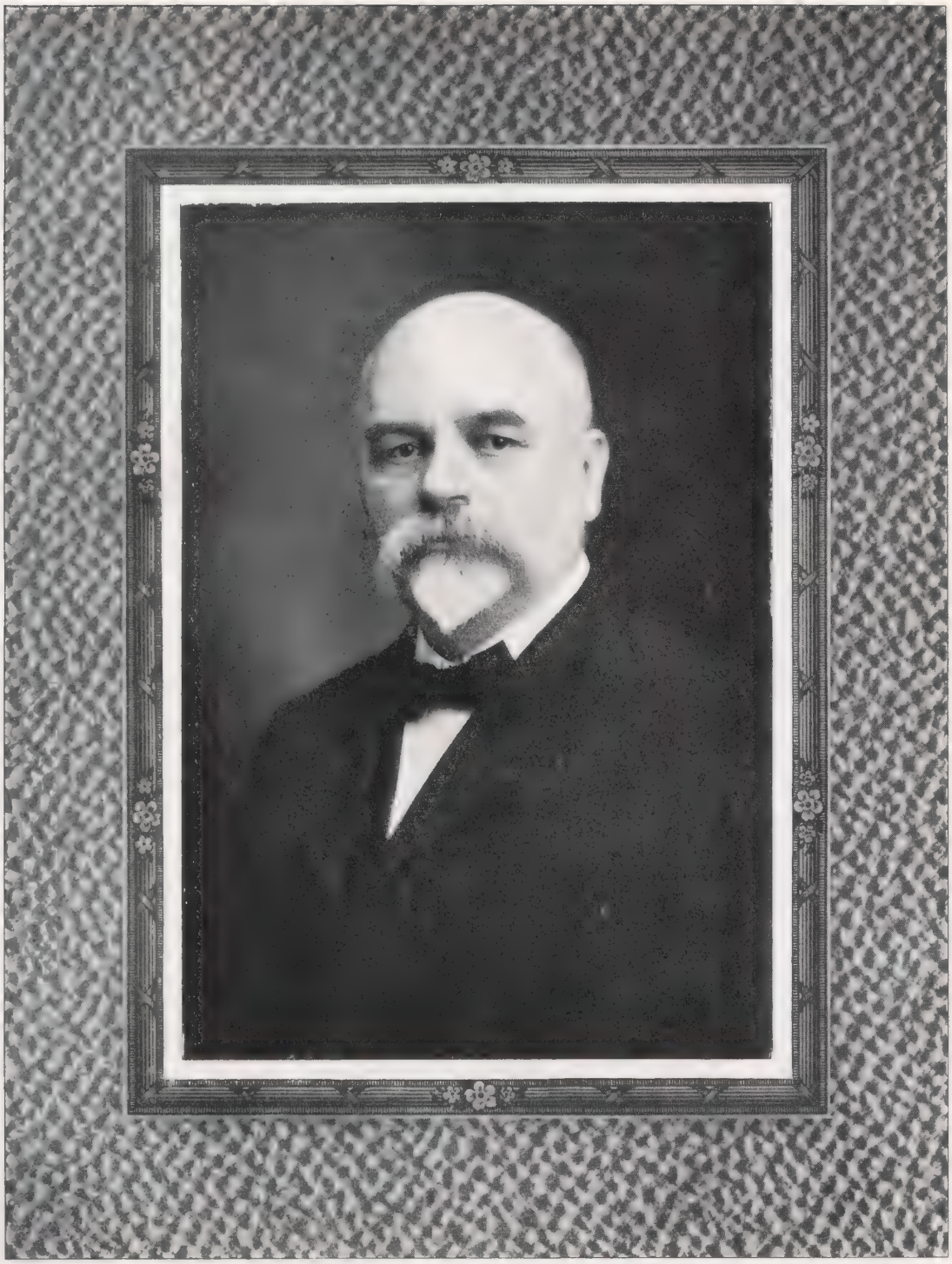


Fig. 1.—General Arrangement of the "Froude."

To obtain comparative data a model of the



PROFESSOR C. H. PEABODY.
Massachusetts Institute of Technology, Boston, U.S.A.

Yours very truly
C. H. Peabody

TABLE I.—WAKE AND THRUST DEDUCTION.

Speed in knots.	Distance of propeller from stern-post.	Revolutions per minute.	Apparent slip.	Shaft horse-power.	$A(1-s)^3$	Real slip.	Wake.	Resistance (pounds).	Thrust (pounds).	Thrust deduction.	Hull efficiency.
7.....	—	351·0	0·1810	8·78	2·89	0·237	0·068	240	370	0·35	0·70
6·5.....	—	315·5	0·1542	6·15	2·80	0·213	0·070	188	294	0·36	0·69
6.....	1″	282·2	0·1263	4·24	2·67	0·186	0·068	147	232	0·37	0·68
5.....	—	225·0	0·0884	2·03	2·53	0·154	0·072	95	144	0·34	0·71
4.....	—	174·8	0·0604	0·92	2·45	0·134	0·078	60	86	0·31	0·70
7.....	—	343·0	0·160	8·50	3·00	0·268	0·129	240	362	0·34	0·76
6·5.....	—	309·0	0.135	6·03	2·91	0·242	0·123	188	286	0·34	0·75
6.....	3″	277·8	0·111	4·24	2·83	0·220	0·123	147	228	0·36	0·73
5.....	—	222·6	0·0791	2·07	2·69	0·187	0·117	95	144	0·34	0·75
4.....	—	174·0	0·0577	0·96	2·59	0·165	0·114	60	86	0·31	0·73
7.....	—	341·5	0·158	7·82	2·78	0·212	0·064	240	294	0·18	0·87
6·5.....	—	305·0	0·126	5·55	2·78	0·211	0·096	188	230	0·18	0·91
6.....	6″	272·5	0·0981	3·85	2·68	0·187	0·099	147	177	0·18	0·92
5.....	—	218·3	0·0603	1·88	2·58	0·165	0·111	95	100	0·05	1·08
4.....	—	172·0	0·0474	0·90	2·52	0·149	0·107	60	54	–0·11	0·99
7.....	—	344·3	0·166	7·66	2·67	0·184	0·022	240	278	0·14	0·88
6·5.....	—	310·5	0·141	5·60	2·66	0·180	0·045	188	214	0·12	0·94
6.....	18″	278·2	0·114	3·94	2·60	0·168	0·061	147	162	0·099	0·96
5.....	—	222·0	0·0767	1·92	2·50	0·146	0·075	95	91	–0·044	1·04
4.....	—	172·0	0·0448	0·90	2·51	0·147	0·107	60	50	–0·020	0·85
7.....	—	338·2	0·151	7·40	2·73	0·196	0·054	240	260	0·077	0·97
6·5.....	—	307·0	0·131	5·41	2·66	0·182	0·058	188	201	0·065	1·00
6.....	30″	277·5	0·112	3·85	2·58	0·164	0·059	147	152	0·040	1·02
5.....	—	224·1	0·083	1·94	2·45	0·138	0·060	95	87	–0·092	0.97
4	—	176·0	0·066	0·89	2·33	0·110	0·047	60	51	–0·176	0·87

TABLE II.—ANALYSIS OF PROGRESSIVE SPEED TRIALS OF THE "MANNING."

1. Speed, knots	7	8	9	10	11	12	13	14	15	16
2. Revolutions per minute	60	68·8	77·4	86·3	95·8	106·2	116·7	127·7	139·5	152·0
3. Indicated horse-power	141	194	263	354	486	671	920	1245	1661	2181
4. Initial friction power	38	44	49	55	61	68	74	81	89	97
5. Load friction power	7	9	12	17	25	35	49	67	91	121
6. Shaft horse-power	96	141	202	282	400	568	797	1097	1481	1963
7. Apparent slip	0·044	0·044	0·045	0·048	0·057	0·071	0·085	0·099	0·117	0·135
8. Real slip	0·11	0·11	0·11	0·11	0·12	0·14	0·15	0·16	0·18	0·20
9. Propeller efficiency	0·68	0·68	0·68	0·68	0·68	0·68	0·68	0·68	0·68	0·68
10. Propeller power	65	96	137	192	272	386	542	746	1000	1334
11. Effective power	58	86	123	173	245	346	488	671	900	1200
12. Surface friction power	52	76	105	142	187	237	299	369	449	539
13. Residual power	6	10	17	31	58	107	189	302	451	661
14. Factor for residual power	—	—	—	—	0·22	0·26	0·32	0·34	0·36	0·38
15. Speed length ratio	—	—	—	—	0·80	0·88	0·95	1·02	1·09	1·17
16. Residual power from 23·5ft. model	—	—	—	—	126	180	271	401	592	850

Manning one-eighth full size (23·5ft. long) was tested in the U.S. model basin at Washington. A model of the *Froude's* propeller was also tested in the Washington tank, the results obtained being used to determine the real slip of the propeller when working astern of the ship so that the wake might be estimated.

From the drawings of the *Manning* it appeared that the propeller was set with its hub 5in. from the sternpost, which corresponded to 1in. for the *Froude*. Tests were also made with the propeller hub successively 3, 6, 18 and 30in. abaft the sternpost. The results (Fig. 2) showed that the location of the propeller in relation to the stern of the boat and the broad sternpost had a large effect on the thrust and power required to drive the boat. The mind at once jumps to the conclusion that a large waste of power is due to setting the propeller in the eddy astern of the rudder post, and that when the propeller is once free from the eddy (as at the 6-in. settings) there is only the normal and regular gain that may be expected from carrying the propeller well away from the hull.

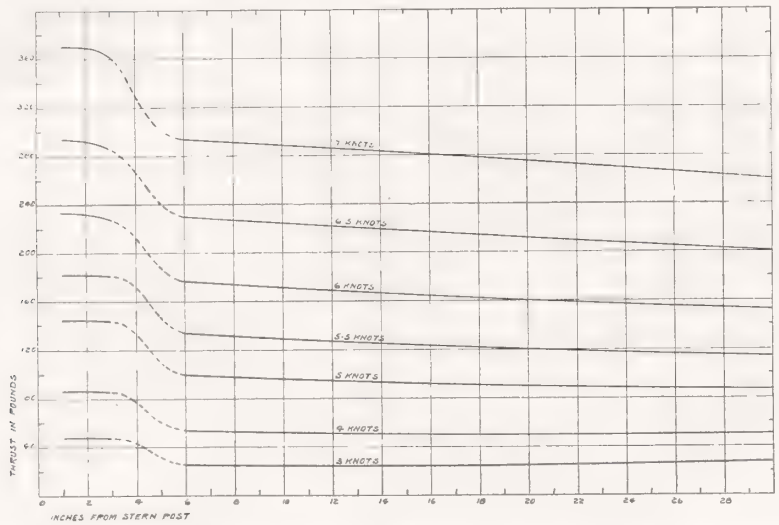


Fig. 2.—Thrust of Propeller at Different Distances from Sternpost.

The results of the investigations of 1910 can be conveniently presented to the mind by computing the wake, the thrust deduction, and the hull efficiency in the conventional way, as given in Table I. The resistance figures given in this table were obtained by towing the *Froude* with the propeller removed, and are shown in Fig. 3, together with results estimated from the 23·5ft. model. The discrepancy between the two curves is not more than may reasonably be charged to the difference in surface. With regard to the degree of precision which can be attributed to the figures in Table I., the author feels justified in claiming accuracy within 1 per cent. for results at full power and speed (6 to 7 knots); at slower speeds and smaller powers the errors are liable to be larger, perhaps as much as 2 per cent. at 4 knots. There are certain anomalies, especially at low powers and speeds, for which no explanation is offered, and which may reveal real anomalies in the accepted conventions concerning the interaction of hull and propeller.

The author has made a new analysis of the trials of the *Manning* compared with his 1899 paper, taking into account the additional information obtained regarding the wake and thrust deduction from the tests of the *Froude*. The new analysis, which is given in Table II., shows appreciable but not large divergence from the earlier one. The discrepancy between the quantities on line 16 and line 13 shows the difference between work with a model and the analysis of a progressive speed trial. From Table II. it will be seen that an estimate of the effective horse-power of the *Manning* at 16 knots, based upon the model tank experiments with the 23·5-ft. model, gives 1,389 horse-power instead of 1,200 deduced from the progressive trials, or an excess of about 15½ per cent. The author enumerates the several elements that may enter into an analysis of this discrepancy as follow :—

(1) The computation of surface friction. All quantities offered for computation for ships are extrapolated from Mr. Wm. Froude's experiments which were for surfaces not over 50ft. long.

(2) The steam engine indicator is notoriously unreliable, 2 to 3 per cent. being a conservative estimate of probable error.

(3) The estimate of hull efficiency from wake and thrust deduction is open to doubt.

(4) Though the theory of mechanical similitude indicates that resistance is proportional to the density of the medium, and therefore it is customary to allow an increase in the ratio of 36 : 35 for ships in sea water, no experimental information on this point is available.

(5) There is reason for questioning the accuracy of the mechanical theory of similitude as applied (a) to the computation of residual resistance and

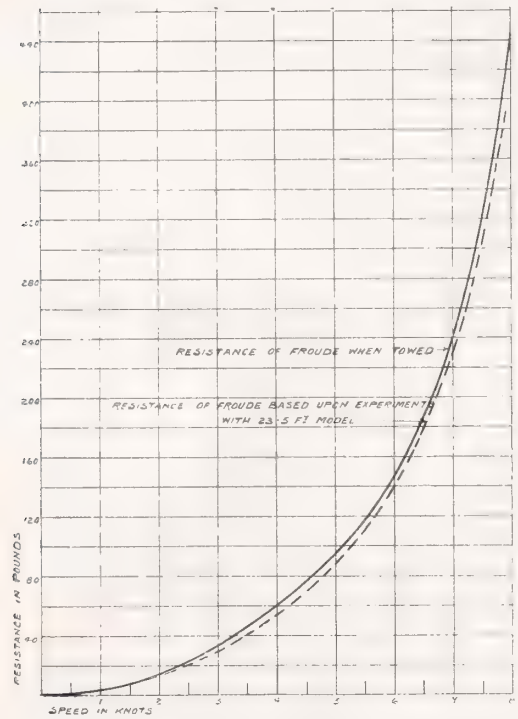

RESISTANCE OF FROUDE WHEN TOWED
RESISTANCE OF FROUDE BASED UPON EXPERIMENTS WITH 23·5 FT MODEL
RESISTANCE IN POUNDS
SPEED IN KNOTS

Fig. 3.—Resistance of the "Froude."

(b) to the properties of propellers, more especially as the pressure of the atmosphere is nearly constant, and does not vary as the theory would indicate.

The main object of the 1911 investigation was the determination of the influence of pitch and area of propeller on propulsion. For this purpose three propellers were made, each 2ft. in diameter and with a projected area of about ·60 of the disc area. Three pitch ratios were chosen, one normal and the other two abnormal, the pitch being 2·2ft., 3ft., and 1·6ft. respectively. After using these propellers at the area ratio given above, they were successively cut down and the backs properly shaped to give projected area-ratios of about ·50 and ·40. Although the tests have not yet been completely analysed for wake, thrust deduction, and hull efficiency, they clearly show that the changes of projected area-ratios and pitch have comparatively small effect on propeller performance ; and as the variation of pitch and width of blades was much larger than is likely to occur in practice, it may be concluded that the practical efficiency of well-designed propellers is very little affected by ordinary variations of pitch and width of blades. To this conclusion it may be added that all propellers having ovoid forms of blades are sensibly equal in efficiency.

APPENDIX.

Method of Allowing for Wind Resistance.

By Professor H. A. Everett.

Let R_w = Wind resistance.

V_w = Speed of wind past the ship.

V'_w = Speed of wind.

V = Speed of ship.

R = Resistance of ship.

When the ship is going against the wind,

$$V_w = V + V'_w$$

and when going with the wind,

$$V_w = V - V'_w.$$

Froude gives for wind resistance,

$$R_w = KAV^2_w$$

where K is a constant and A is the area of the transverse contour above the L. W. L.

$$\therefore \quad R_w \propto V^2_w$$

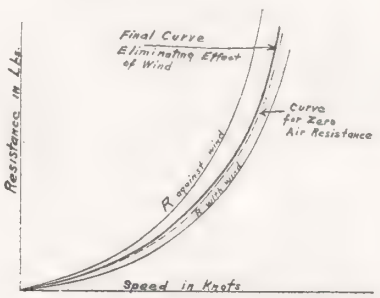


Fig. 4.—Effect of Wind.

From the experimental curves we get R_w and R_a at any desired speed of the ship V.

Then—

$$R_{\text{against}} = R + R_w \text{ and} \quad (1)$$

$$R_{\text{with}} = R - R_w \quad (2)$$

$$R_{\text{against}} = R + KA\,(V + V'_w)^2 \quad (3)$$

$$R_{\text{with}} = R - KA\,(V - V'_w)^2 \quad (4)$$

$$R_{\text{against}} - R_{\text{with}} = KA\{(V + V'_w)^2 + (V - V'_w)^2\}$$

$$\frac{R_{\text{against}} - R_{\text{with}}}{KA} = (V + V'_w)^2 + (V - V'_w)^2$$

Completing the square gives,

$$\frac{R_{\text{against}} - R_{\text{with}}}{KA} + 2\,(V + V'_w)\,(V - V'_w) = \{(V + V'_w) + (V - V'_w)\}^2$$

$$\frac{R_{\text{against}} - R_{\text{with}}}{KA} + 2\,V^2 - 2\,V'^2_w = (2\,V)^2 = 4\,V^2$$

$$\frac{R_{\text{against}} - R_{\text{with}}}{KA} - 4\,V^2 + 2\,V^2 = 2\,V'^2_w$$

$$V'_w = \sqrt{\frac{R_{\text{against}} - R_{\text{with}}}{2\,KA}} - V_2$$

To illustrate the use of this method, in the case of the experimental model *Froude*, $2\,KA = \cdot 34$.

On the runs of the 23rd August at 7 knots the thrust with the wind was 286 pounds and against the wind was 322 pounds. Whence,

$$V'_w = \sqrt{\frac{322 - 286}{\cdot 34}} - 49 = \sqrt{\frac{36}{\cdot 34}} - 49 = 7 \cdot 5.$$

Therefore the speed of the wind $V'_w = 7 \cdot 5$ knots.

With the velocity of the wind known it is now possible by using Froude's formula to figure the wind resistance of the boat at any speed in either direction, as for instance at 7 knots.

$$R_{w\,\text{against}} = 0 \cdot 17\ (14 \cdot 5)^2 = 35 \cdot 7$$

$$R_{w\,\text{with}} = 0 \cdot 17\ (\ 0 \cdot 5)^2 = \ 0 \cdot 4.$$

Now the total divergence between the two curves at 7 knots should be equal to the sum of these quantities.

$$35 \cdot 7 + 0 \cdot 4 = 36 \cdot 1 \text{ pounds.}$$

As a matter of fact, the difference was actually 36 pounds.

Subtracting the wind resistance from the curve with the wind, will give a curve for zero air resistance ; that is for a ship always accompanied by air going at the same velocity that the ship is.

Now to the curve of zero air resistance should be added the normal air resistance due to the speed of the ship through the still air which gives the heavy full line curve of Fig. 4, and this is a curve from which wind influence has been eliminated.

THE EFFECT OF WAVES UPON A TAFFRAIL LOG.

By Professor Harold A. Everett.

(*American Society of Naval Architects and Marine Engineers ; November*, 1911.)

TAFFRAIL logs have had, in general, a bad name for accuracy. The causes of their erratic behaviour have been variously ascribed to poor mechanical construction, wake, and torsional elasticity of the log-line. In the case of a log used by the author great care was taken to eliminate the first and last sources of error ; and in order to do away with the second, the log was towed from the end of a 25-ft.boom. As the erratic behaviour was still found to exist, the author was led to suspect that the log was affected by the waves which the ship carries along with herself, and to consider what effect upon such a log might be expected from these waves.

Fig. 1 shows the profile of a trochoidal wave travelling across the page from right to left. This wave is composed of particles, each rotating in a circular orbit, and the orbits of three of these particles are shown. If a log were to be dragged

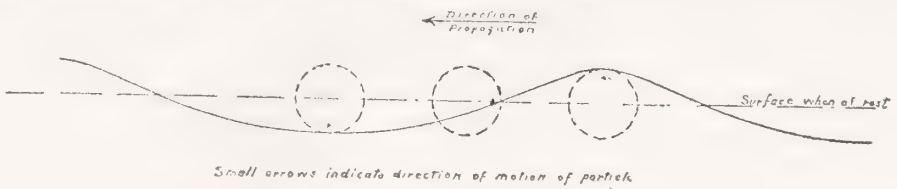


Fig. 1.—Trochoidal Wave.

in this wave at the same speed as the wave travels, and from a line of such length that it came at a hollow, and therefore among particles which are moving away from the ship, then the rotator might be expected to run faster than it would in still water. If the line were of such a length that the log came at a wave crest, then the opposite effect might be expected. All ships when underway maintain two systems of waves, a bow system and a stern system, each consisting of diagonal waves on each side, with transverse waves which terminate in these diagonal waves as the ship increases her speed. The transverse waves gradually lengthen; and the bow wave system, when it reaches the stern, alternately increases and decreases the height of the resultant stern system.

The log will, therefore, be differently situated in relation to a wave crest or trough according to the rate at which the vessel is travelling, and will underrun or overrun as the case may be. The wave lengths at different speeds of the ship can be figured with reasonable exactness from theoretical considerations, which have been verified by experimental work in towing basins and elsewhere, and, as thus determined, are shown in Fig. 2 for the U.S. Revenue Cutter *Gresham*, a vessel 188ft. long. On the trials of this vessel the author arranged to measure speeds by log which could be compared with the speeds as determined by runs over a measured course.

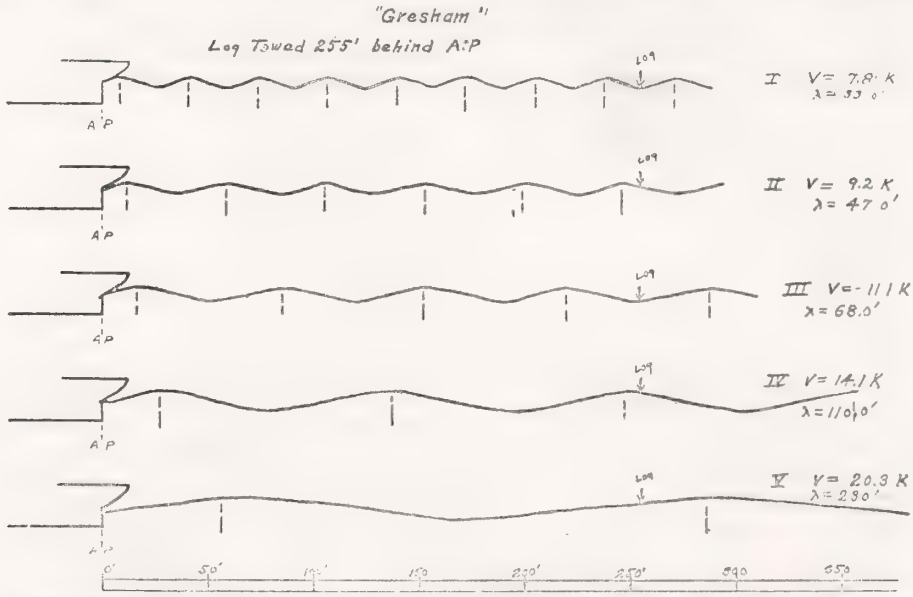


Fig. 2.—"Gresham": Stern Wave System at Speeds at which Bow Wave System superposes.

If the log factor for different speeds in water having no wave motion is represented by the uniform curve *AB* (Fig. 3), then the log factors which will give the true speeds in the case of a log towed 255ft. behind a vessel 188ft. long should lie on an undulating curve, shown dotted in Fig. 3, in which the variations to be expected are set off from *AB* as a base line. From Figs. 2 and 3 it will be seen that, on the basis of the author's theory, at speeds of 14·1, 11·1, 9·2 and 7·8 knots the variation of the log factor from the

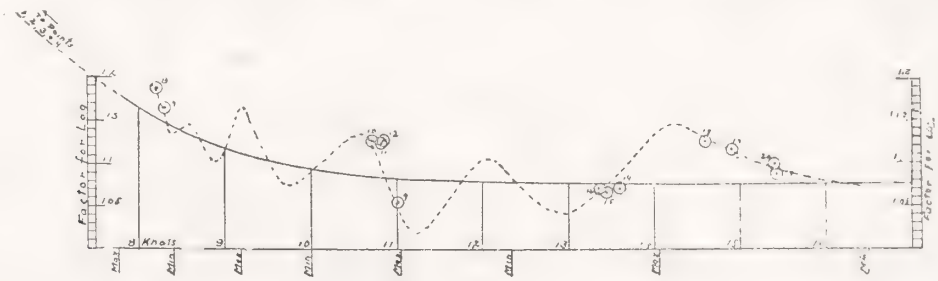


Fig. 3.—"Gresham": Log Factors.

still-water line *AB* should be greatest, and at speeds of 16·5, 12·35, 10 and 8·35 knots the variation should be small. The actual log factors, as deducted from the trials of the *Gresham* by expressing the speeds as determined by log in relation to the actual speeds, are plotted in Fig. 3 and lie very nearly on the undulating curve. The results were confirmed by a similar investigation conducted at the progressive trials of the steamer *Sankaty*.

Professor Everett's investigations show definitely :—

1.—Any taffrail log is seriously affected by the wave system carried along by the ship.

2.—In order to be available for measurement of distances at different speeds, the log must be calibrated by obtaining many points at closely adjacent speeds, and in some manner that takes account of the wave action.

3.—The length of the log-line must be unchanged throughout calibration and subsequent running.

4.—A taffrail log as a close measurer of speeds is a questionable instrument.

THE RESISTANCE OF SOME MERCHANT SHIP TYPES IN SHALLOW WATER.

By Professor H. C. Sadler.

(*American Society of Naval Architects and Marine Engineers ; November*, 1911.)

THE author presents certain experimental results which have been obtained with a number of models of various merchant ship types when tested in water of varying depth in the experimental tank at the University of Michigan. The models tried ranged from fine to full types, but it is not claimed that the whole field has been covered. For each model curves are given showing the residuary resistance in pounds per ton of displacement at different speeds and depths of water, similar to that reproduced in Fig. 1. It is of interest to note that the "humps" in the resistance curves occurred in all cases at practically the same speed and depth of water, irrespective of the lengths of the models, which indicates that the speed at which the maximum residuary resistance occurs is a function of depth of water rather than

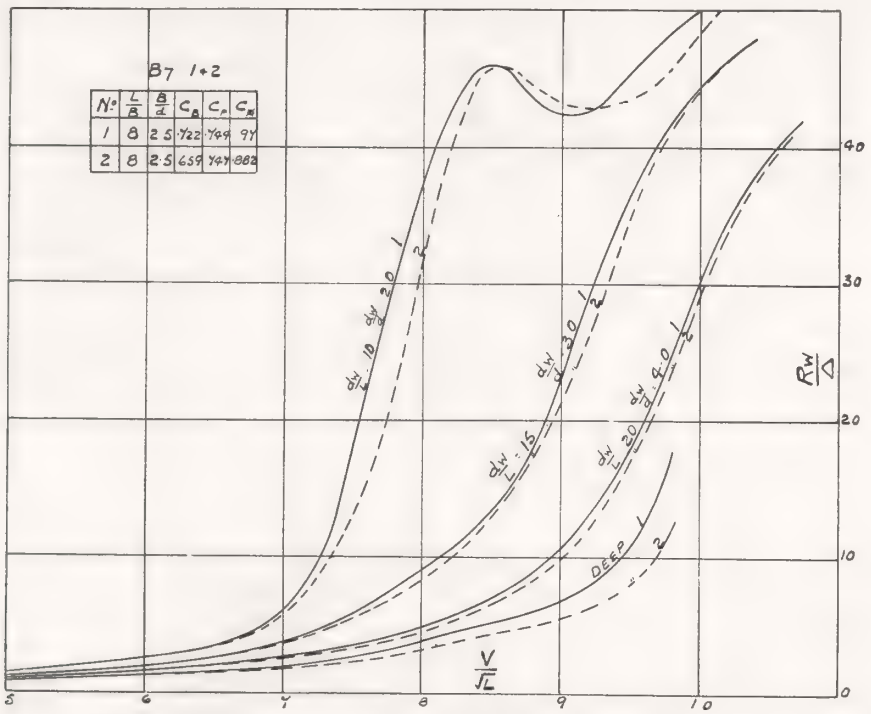


Fig. 1.—Curves in Residuary Resistance in Shallow Water.

size of ship. There is, however, a tendency for the maximum to occur at a higher speed as the form of the vessel becomes finer, as is illustrated by the curves for the two models given in Fig. 1. It is often of interest to know at about what speed the first "hump" is likely to occur for a given depth of water. The speed-length ratios at which this maximum resistance may be expected, from the author's experiments, have been plotted on a depth-length ratio base in Fig. 2. In this case the total resistance of the model has been taken. Other published results have been added for comparison; and from these, which are mostly for torpedo-boat destroyers and vessels of a somewhat similar type, it appears as though the "hump" for a given depth of water occurs at slightly higher speed in the finer forms. Observations of the wave formation at the critical speeds confirm those already given by previous writers. At or near these speeds the vessel tends to form a train of waves at the stern, which extends for some distance from each side of the ship and at right angles to the same. After passing this point the normal type of bow and stern waves gradually re-appear.

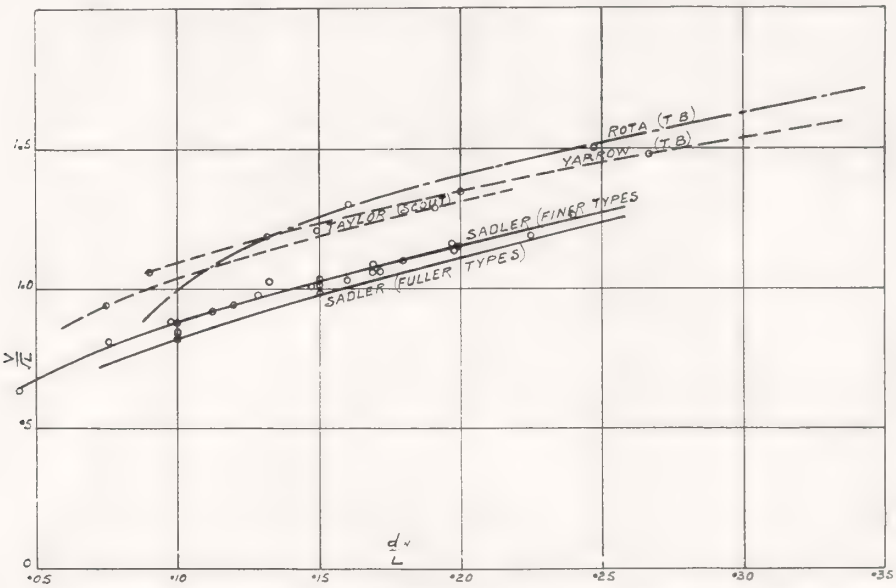


Fig. 2.—Curves of Critical Speed in Relation to Depth of Water.

THE WILLIAM FROUDE NATIONAL TANK.

By Mr. G. S. Baker.

(*Institution of Naval Architects; March*, 1912.)

MR. Baker in his paper describes the smaller tank which has been built as an annexe to the William Froude Experimental Tank at the National Physical Laboratory. The dimensions of this tank are as follow:—

Total length of water surface	64·5 ft.
Breadth at the uniform section	5·0 „
Depth at the uniform section	3·12 „

The design and equipment of the tank will be seen from Fig. 1, arrangements being provided both for still and flowing water experiments. The tank is intended for general laboratory experiments and for more or less exploratory work with models. The errors which may creep into the results obtained with ship models in such a small tank are very numerous. Those of the most serious nature are due to inertia and the way in which the value of surface friction varies with length.

In order to test the degree of accuracy obtainable with a model of ordinary form in the small tank, models 3ft., 4ft., and 15ft. were made, all of exactly the same shape. The 15-ft. model was run in the main tank and the other two in the small tank, and the results for the latter were corrected for skin friction to a 15-ft. length by

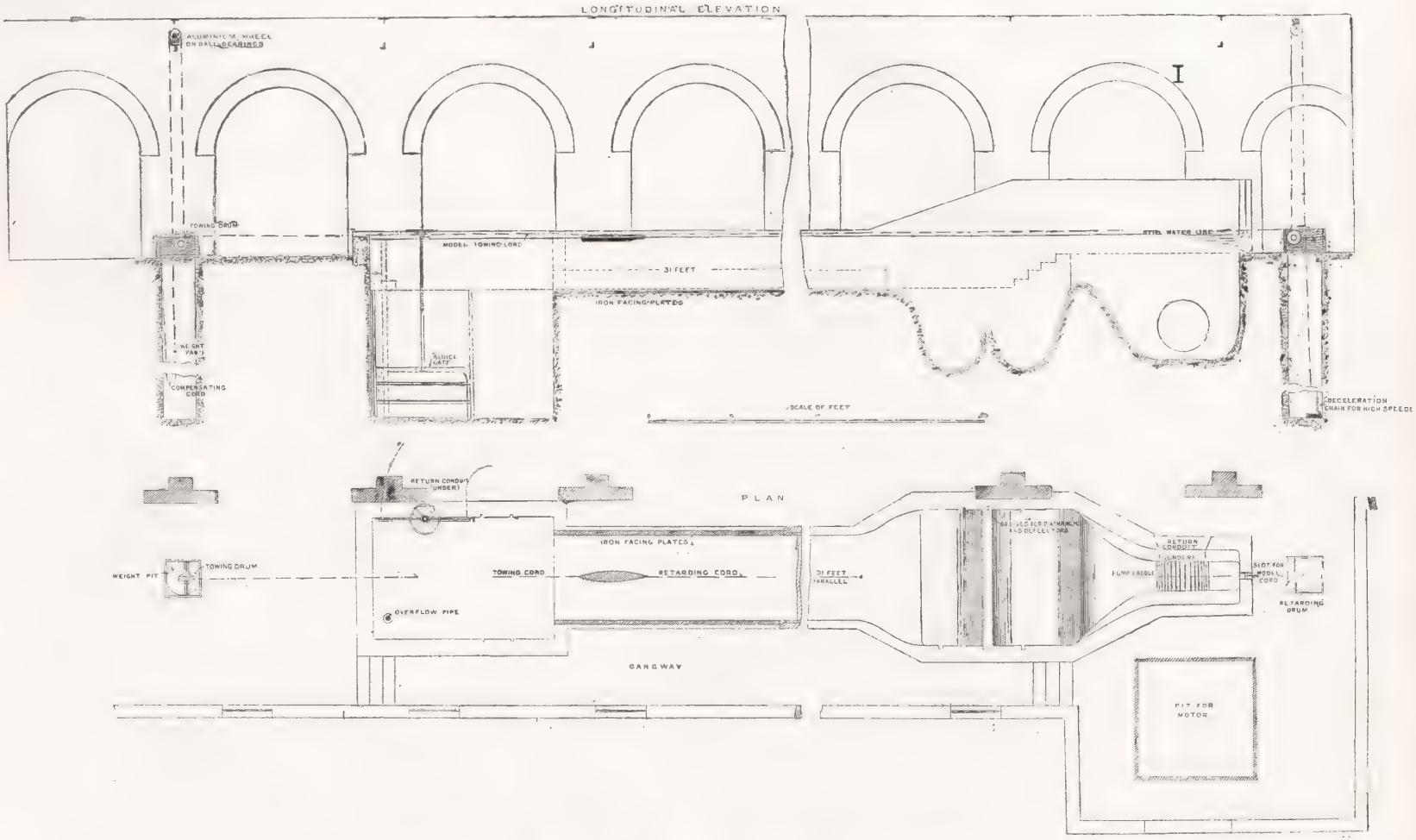


Fig. 1.—The Small Tank at the National Physical Laboratory.

means of the curves of frictional resistance of planks obtained by the late Mr. W. Froude. At a value of $\frac{V}{\sqrt{L}}$ equal to unity, the 3-ft. model results (corrected for friction of apparatus) varied some 13 per cent. for different experiments on different days, and the 4-ft. model results varied 6 per cent. At $\frac{V}{\sqrt{L}}$ equal to two, the corresponding figures were approximately $5\frac{1}{2}$ per cent. and 4 per cent. respectively. Below a value of $\frac{V}{\sqrt{L}}$ equal to unity, the 4-ft. model results fell away some 8 per cent. compared with the large model figures; between $\frac{V}{\sqrt{L}}$ equal to 1·1 and 1·5 the curves for the two models agreed fairly well; and for higher speeds the 4-ft. model results again fell away slowly, the difference being 8 per cent. at $\frac{V}{\sqrt{L}}$ equal to 1·7. The experiments on which the above figures are based were made on different days over a period of a month, and when an opportunity occurs similar experiments will be made with a much less resistful form.

THE LAW OF COMPARISON FOR SURFACE FRICTION AND EDDY-MAKING RESISTANCES IN FLUIDS.

By Mr. T. E. Stanton, D.Sc.

(*Institution of Naval Architects; March*, 1912.)

THE discovery of a law of comparison for surface friction was made by Osborne Reynolds nearly thirty years ago. What Reynolds found was that in the flow of water through parallel pipes steady motion broke down and eddying began at a certain critical value of the velocity, which varied inversely with the diameter of the pipe. In other words, eddying began at a definite value of $V \times L$, where V is the velocity and L the diameter of the pipe. He then proceeded to the determination of the frictional resistance of the surfaces of pipes and found that at all speeds, whether above or below the critical speed, if the velocities of the various pipes were so regulated that VL had the same value for each pipe, the frictional resistances per unit area of the surfaces were in the ratio of the squares of the corresponding speeds. Expressed in a simpler way still, the results were that in two pipes in which the lengths, diameters, and surface irregularities were in a given constant ratio to each other, if the velocities were made in the inverse of this ratio, the total frictional force was the same for each pipe. This statement assumes that the viscosity of the fluid was the same in both pipes. The full statement of the law is that for similarity of motion the quantity $\rho\frac{vl}{\mu}$ (where μ is the coefficient of viscosity of the fluid, and ρ the density) must be the same in each case.

The resistance of a body of any shape moving in a fluid appears to depend solely on the dimensions and nature of surface of the body and on the velocity, density, and viscosity of the fluid, provided:—

1.—That the immersion is of such a depth that there is no resistance due to the formation of surface waves.

2.—That the velocities are not so high as to cause cavitation or serious changes of density.

In such a case, by the application of the principle of dynamical similarity, Lord Rayleigh* has shown that the resistance per unit area can be expressed by

$$R = \tfrac{1}{2}\,\rho V^2\, k$$

where k is an expression depending solely on the value of $\rho\frac{vl}{\mu}$. It will be seen at once that Reynolds' Law of Comparison for friction is only a particular case of this general relation when the motion is tangential to the surface of the body; and, further, that this law of comparison will apply equally well to cases in which the resistance is made up partly of surface friction and partly of eddy-making.

Experimental verifications of the truth of this relation have recently been obtained at the National Physical Laboratory in the two following cases:—

1.—The determination of the frictional resistance of two different fluids (air and water) on the same surface.

2.—The total resistance in water of models of a dirigible balloon made to different scales.

In the case of the experiments with balloon models, one model was made 12in. diameter and

* Report of Advisory Committee of Aëronautics; Vol. I., page 38.

the other 4in. diameter. The results obtained are given in Table I.

TABLE I.

Value of $V.L$, ft.²/sec.	Resistance in lb.	
	Model 12in. diameter.	Model 4in. diameter.
3·16	0·45	0·46
3·87	0·63	0·65
4·47	0·85	0·84
5·00	1·05	1·02
5·48	1·28	1·22

At speeds of and above 5ft. per second the larger model was not immersed to sufficiently great a depth to avoid wave-making, which accounts for its resistance being appreciably above that of the smaller model in the last two results given.

As regards the practical application of this law of comparison, the cases in which its use is valuable are the predictions of the resistances of full-sized dirigible balloons, submarines, and torpedoes, for which no other method of equal accuracy is available. For example, to find the resistance of a dirigible balloon, moving at V ft. per sec., a scale model is made and its resistance determined in a water tank. If L and l are the dimensions of the balloon and model, then since for comparison the values of $\rho\frac{VL}{\mu}$ must be the same for each, and the value of $\frac{\mu}{\rho}$ for air is approximately 13 times the corresponding value for water, the speed v at which the model must be towed is $\frac{VL}{13l}$. Then, if R is the resistance of the model at this speed, the resistance of the full-sized dirigible at V ft. per second will be given by :—

$$R\frac{\rho_A}{\rho_w}\left(\frac{VL}{vl}\right)^2 = \frac{R}{4{\cdot}7} \text{ approximately,}$$

where ρ_A is the density of air and ρ_w is the density of water.

As regards the calculations of the frictional and eddy-making resistances of ships, any practical application of this law is at once seen to be out of the question, owing to the enormous speeds at which it would be necessary to run the model, and all that could be done would be a determination of the true wave-making resistance of the model as distinguished from the frictional and eddy-making resistances.

In conclusion Dr. Stanton points out that from Lord Rayleigh's equation the eddy-making resistance is not exactly proportional to the square of the speed, which will make the application of Froude's Law of Comparison to eddy-making not strictly correct, although, of course, the error involved must necessarily be a small one.

THE WORK OF THE EXPERIMENTAL TANK AT THE ARSENAL OF SPEZIA FROM 1889 TO 1910.

By Ing. Com. G. ROTA, Colonel, Italian Corps of Naval Constructors.

(*Collegio degli Ingegneri Navali Italiani ; November*, 1911.)

THE Spezia tank was founded in 1889 owing to the initiative of Benedetto Brin, and was the third model experimental tank constructed in order of time, following the tank erected for the British Admiralty at Gosport and that for Messrs. Denny at Dumbarton. The great utility of the tank was soon demonstrated, and since 1891 models of all war vessels for the Italian Navy have been tested to determine the best form to adopt, and the resistance, speed and power associated with that form.

From the commencement the results of the model experiments have been expressed, independent of scale, on the Constant System of Notation devised by Mr. R. E. Froude. Another method used at Spezia for standardizing results, which is very useful for preliminary designing work, is the employment of a logarithmic chart in the manner first suggested by Mr. Denholm Young. The form of each model is also standardized, independent of scale, on a diagram giving the curve of sectional areas, midship section, and load waterline, lineal dimensions being expressed by the ratio $\frac{l}{d^{\frac{1}{3}}}$ and areas by $\frac{s}{d^{\frac{2}{3}}}$ where l is the dimension, s the area, and d the displacement.

In addition to determining the resistance and the trim of the models at various speeds, it has

from the first been the custom to photograph the wave formations on the side of the model, the position of the wave waterline being identified by horizontal and transverse lines drawn on the model. Observations of the actual wave profiles on the trial trips of ships have shown for corresponding speeds perfect agreement with the waves observed at the tank experiments with their models.

In 1899 a series of original experiments were carried out to determine the effect on resistance of alterations in length, breadth or draught, leaving in every case two dimensions unaltered. It was found :—

1.—An increase or diminution of breadth or draught (all dimensions in the direction of breadth or draught being, of course, varied in like manner) causes an increase or diminution in the E.H.P. necessary for a given speed.

2.—For an equal alteration in displacement the increase or decrease of E.H.P. is greater for an alteration of breadth than for an alteration of draught.

3.—For a given variation in displacement the alteration in the resistance due to skin friction is the same, whether the change in displacement is effected by altering the breadth or the draught.

4.—Increase of length is usually associated with a marked diminution in wave-making resistance, but, on the other hand, the skin friction is increased.

5.—Every length of ship has a particular speed at which alterations in displacement (within ordinary limits) have little effect upon the power producing that speed. The relationship between length and speed at which this occurs is given approximately by $V = \sqrt{L}$, where V is the speed in knots and L the length in feet.

6.—In the case of a model of given length and speed, small alterations in the displacement, if effected by alterations in breadth, alter the E.H.P. in practically the same ratio as the displacement has altered. If, however, the change in displacement is effected by change in draught, the alteration in E.H.P. is only about two-thirds as much. With regard to the effect of change of length, for a given speed $V = \sqrt{L}$ alterations of the original length will cause an increase in power whether the displacement be increased or diminished. If $V > \sqrt{L}$ then an increase in displacement will be associated with a decrease in power, and a decrease in displacement with an increase in power. If $V < \sqrt{L}$ the converse is the case.

A series of interesting experiments were carried out to investigate the influence of change on the ratio of beam to draught in models having the same lengths, displacements, and curves of sectional areas. It was found that the effect upon the E.H.P. was mainly that due to change in the skin friction, the minimum resistance corresponding to the model with the minimum wetted surface. Considerable variation, therefore, can be made in the ratio of beam to draught without any great effect upon the power required.

In 1889 a series of rolling experiments were begun with small models to ascertain in a general way the effect of bilge keels upon the rolling of ships. Each model used was carefully made to correspond to an actual ship as regards displacement, centre of gravity, and moment of inertia, and to have a period of roll corresponding to that of the ship, such that $\frac{T}{t} = \left(\frac{D}{d}\right)^{\frac{1}{6}}$, T and t being the periods of roll and D and d the displacements of the ship and model respectively. The curves of extinction obtained with the models were compared with those obtained with the actual ships at sea, and perfect agreement was observed.

In 1891 the apparatus for experimenting with model propellers was acquired, and a commencement made with the study of the efficiency of propellers when towed alone and when towed behind ship models. For tabulating the results the method of Constants suggested by Mr. R. E. Froude was adopted, with the exception that instead of using the values n and s, number of revolutions and thrust of propeller, the following were used, *i.e.*, $N = \frac{nd}{v}$ and $S = \frac{s}{d^2 v^2}$, where d is the diameter of the propeller and v the lineal speed of advance. With this apparatus much valuable information was accumulated with regard to propellers of many types, varied pitch, and working with different slips. The law of similarity was repeatedly verified by testing models of the same propellers on different scales, when the results expressed on the Constant system, independent of scale, were found to agree exactly.

Particular study of the pitch of propellers was taken up in 1896, and a rational notion of pitch was conceived, which was afterwards accepted and used by Mr. R. E. Froude in his 1908 paper contributed to the Institution of Naval Architects. In the system of analysis described in that paper the pitch P is throughout taken as equal to the travel per revolution for zero thrust.

Experiments were carried out to find the influence of depth of immersion upon propellers. It was found that :—

1.—For equal increase of depth, the increase of thrust is greater as the number of revolutions is greater.

2.—The thrust becomes practically constant for the same number of revolutions from a limited to an increased depth of immersion.

3.—The efficiency falls off in a marked way if the blades are not fully immersed.

4.—In the case of fully immersed propellers the efficiency increases with increase in depth of immersion.

In 1904, a special study was made to ascertain the importance of the thickness of propeller blades, the propellers tested being otherwise similar as regards surface. An increase in thickness of the blades was observed to cause a sensible diminution in efficiency. In round numbers, an increase of 30 per cent. in thickness caused a reduction in the efficiency of 11 per cent.

It is hardly necessary to point out the great utility of the experiments which can be carried out in an experimental tank with model screws placed behind hull models. Apart from the primary object of determining the resistance and speed of the model, and the number of revolutions of the propeller under conditions which truly represent the conditions in the case of the actual ship, such experiments provide a solution of numerous problems with regard to the number and positions of the propellers.

The dynamometric apparatus of the tank was used to investigate the question of the direction of rotation of twin screws, *i.e.*, whether they should turn inwards or outwards. It was found, broadly, that from the point of view of propulsive efficiency it is practically immaterial whether the screws rotate one way or the other, provided they are placed well aft and clear of the hull. When, however, the screws are placed more forward, there is a loss of efficiency if they rotate inwards.

When the turbine was first introduced for ship propulsion, it was suggested that the adoption of several propellers on one turbine-driven shaft would be advantageous. Such an arrangement was tried in 1897 at the Spezia tank, but was proved to be not worth adopting, as it showed a loss in efficiency up to 10 per cent. On the other hand, contrary turning screws on a common axis, actuated by independent concentric shafts, gave good results, showing a gain in efficiency of from 15 to 20 per cent. The Spezia experiments were confirmed by actual trials made with a steam launch at the Dockyard of Castellamare, which was fitted with double contrary-turning propellers on the same shaft which had previously driven an ordinary single screw. The comparative results showed in this case a gain of 20 per cent. in efficiency due to the double-screw arrangement.

Two years ago an apparatus was devised and constructed in the workshop of the tank to experiment with quadruple screws, each on a separate shaft and disposed in the manner which has been adopted in a number of turbine-driven warships. Experiments with this apparatus showed that propulsion by four screws is less efficient than by twin screws. The inner screws were found to be from 20 per cent. to 17 per cent., according to speed, more efficient than the outer screws. There was also a gain in the wake coefficient at the inner screws of about 7 to 8 per cent.

In 1898 systematic experiments were carried out to determine the influence of depth of water upon speed, which, at that time, was an obscure problem in connection with speed trials. The results were tabulated in the form of curves, from which it is possible to estimate the depth necessary to avoid shallow-water effects in vessels of varied form and displacement. The results obtained at the Spezia tank have since been confirmed by numerous other experimenters.

In 1900 experiments were made to test the towing of barges for canal service, the trials being carried out in a miniature canal formed by wood flooring and sides inside the main tank. The sides were arranged so that they could be held either in a vertical or inclined position. The results made it very clear that the trapezoidal form of canal is to be preferred to the rectangular one.

Novel experiments are now being carried out at the Spezia tank to determine the efficiency of rudders. The investigations will comprise the questions of position of the rudder, rudder area, power applied at the tiller, and turning circles, and, it is hoped, will prove of great value to naval architects.

In addition to the work carried out for the Italian naval authorities, numerous models have been tested at Spezia for Italian shipbuilding firms and also for foreign naval authorities. From his brief survey of the work accomplished, the author thinks it will be conceded that results achieved have fully justified the outlay of the sum of £6,000 originally expended in founding the tank.

THE EXPERIMENTAL STUDY OF SHIP RESISTANCE.

By Professor Angelo Scribanti.

(*Collegio degli Ingegneri Navali Italiani; November*, 1911.)

PROFESSOR Scribanti summarizes, in this very concise and carefully written work, the results of model experiments of various types of vessels employed in the merchant and warship services. The results have all been reduced to a standard displacement of one kilogramme by the application of the law of corresponding speeds. By this law, if d is the displacement, r the resistance, and v the speed of the model, the relationship between x and y is independent of scale when

$$x = \frac{v}{d^{\frac{1}{6}}} \text{ and } y = \frac{r}{d}$$

To fully define the results, a coefficient of form is required, and for this purpose the author uses the ratio $z = \frac{l}{d^{\frac{1}{3}}}$ where l is the length of the model. If the values of x, y and z are associated with a displacement of one kilogramme, then for a model reduced to this displacement the length is z decimetres and the resistance y grammes at a speed of x metres per second. The method permits a ready comparison of the resistance of different types of vessels, such as shown for different types of merchant vessels in Fig. 1 and different classes of warships in Fig. 2, while 22 similar diagrams are given dealing with all types of models in greater detail. The author has also standardized on this system the experiments made on the influence of depth of water on resistance by various experimenters, the results being given in a further set of 15 diagrams attached to the paper.

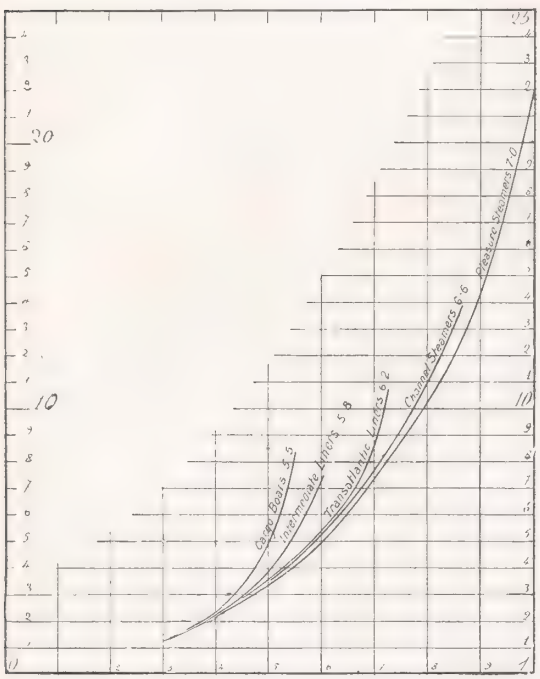


Fig. 1.—Resistance of Different Types of Merchant Vessels.

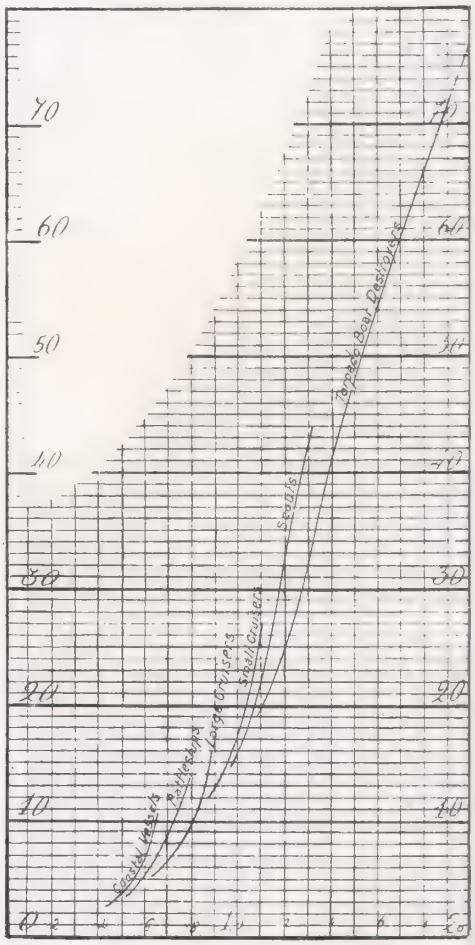


Fig. 2.—Resistance of Different Types of Warships.

THE RESISTANCE TO MOTION OF SUBMERGED SUBMARINE BOATS.

By Ing. Leonardo Fea, Captain, Italian Corps of Naval Constructors.

(*Collegio degli Ingegneri Navali Italiani; November*, 1911.)

THIS short paper deals with the variation of resistance to motion of submarines at different depths of immersion, and the influence of appendages upon the resistance of submarines as determined by model experiments at the experimental tank of Spezia. The model with which the experiments were made had the following characteristics :—

$$\frac{\text{Length}}{\text{Displacement}^{\frac{1}{3}}} = 5{\cdot}595$$

$$\frac{\text{Breadth}}{\text{Displacement}^{\frac{1}{3}}} = {\cdot}527$$

$$\frac{\text{Sectional area}}{\text{Displacement}^{\frac{2}{3}}} = {\cdot}305$$

The results for two of the depths tried are given in Fig. 1 and illustrate the general phenomena observed. This diagram shows the effective horse-power required for a vessel of 500 metrical tons displacement at different speeds with and without appendages, and at depths of immersion for this size of ship of 3·65 and 4·70 metres.

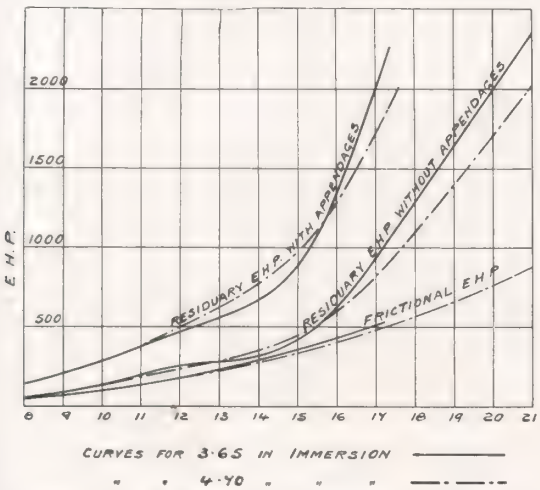


Fig. 1.—Resistance of Submarines at two Different Depths of Immersion.

The powers absorbed in skin friction and wave-making are shown separately in Fig. 1, and their values at any speed must be added together to give the total E.H.P. at that speed. The relative effect upon the E.H.P. absorbed in wave-making of increasing the depth from 3·65 to 4·70 m. in the case of the model with appendages is shown in Table I.

Table I.

Speed in Knots.	E.H.P. 3·65m. immersion. (1)	E.H.P. 4·70m. immersion. (2)	Difference. $\frac{(1)-(2)}{(1)}$ per cent.
10	283	283	0
12	483	514	− 6·4
14	667	782	−17·3
16	1,329	1,292	+ 2·8
17	2,019	1,722	+14·7

The relative effect of the appendages upon the total E.H.P. is shown, for a depth of immersion of 3·65m., in Table II.

Table II.

Speed in Knots.	E.H.P. Naked Model.	E.H.P. Model with Appendages.	Increase in E.H.P. per cent.
8	131	208	56
10	250	396	58
12	460	672	46
14	625	960	54
16	1,056	1,760	66
18	1,875	3,320	77

THE NUMBER OF PROPELLERS IN MODERN WARSHIPS.

By Ing. Nino Pecoraro, Major, Italian Corps of Naval Constructors.

(*Collegio degli Ingegneri Navali Italiani; November*, 1911.)

THE author records some interesting experiments made at the Spezia experimental tank to determine the relative efficiency of a triple-screw arrangement compared with a quadruple-screw arrangement for a modern battleship. Two models were tried, each with

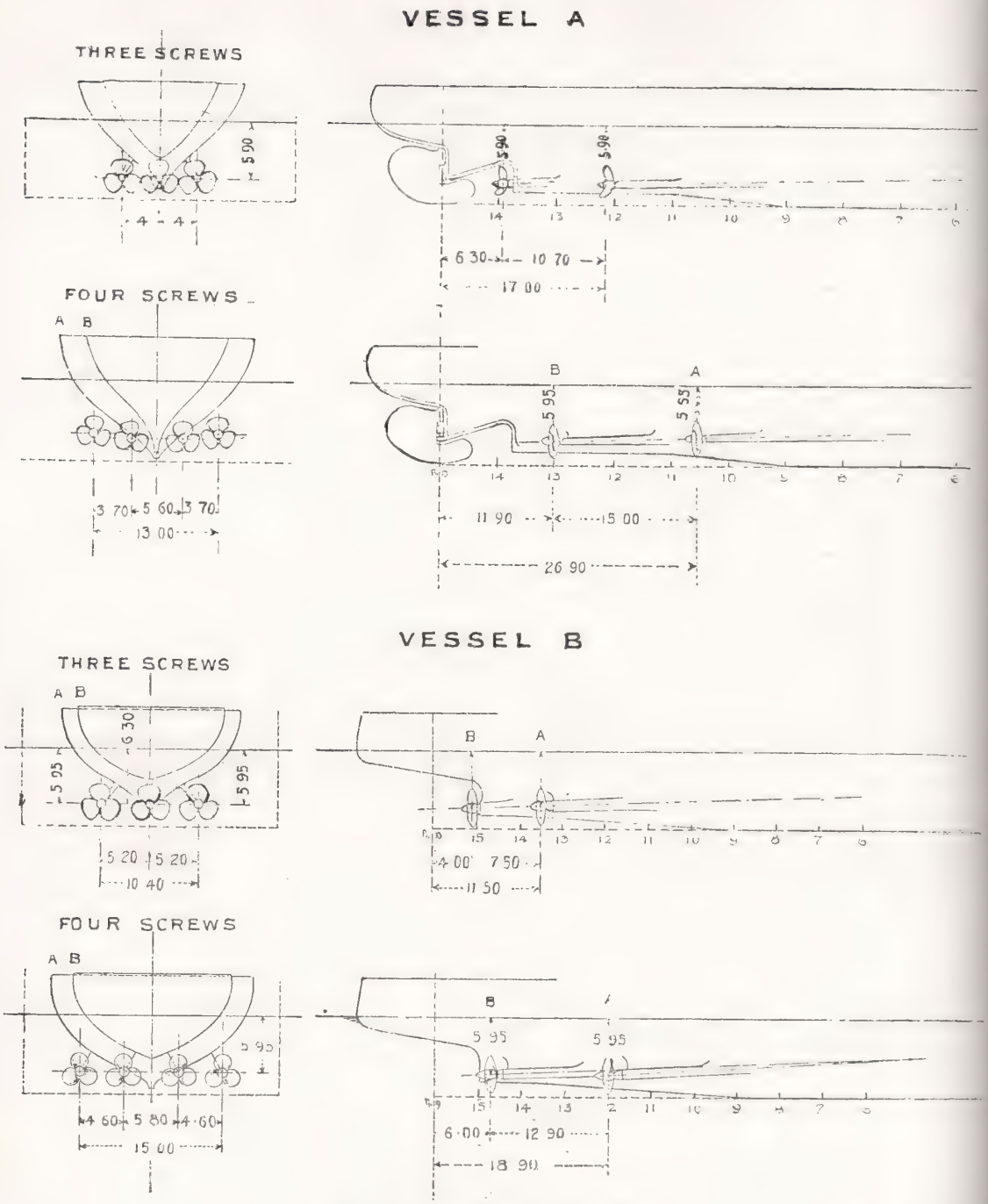


Fig. 1.—Disposition of Propellers.

three and four propellers. The models represented vessels having the following principal dimensions :—

	Vessel A.	Vessel B.
Length B.P., metres	192·00	189·00
Breadth, extreme, metres	28·40	27·40
Mean draught, metres	8·40	8·50
Displacement, metrical tons	24,500	24,400

The form of stern and disposition of propellers in each case are shown in Fig. 1. The models were towed bare and with the appendages attached. The propeller particulars are given in Table I. Tank experiments with models of the various screws showed their efficiency, without the ship model in front, to be practically equal.

For the propellers working behind the ship the efficiency of propulsion $E = \frac{r}{r_1} \times \frac{r_1}{s} \times \frac{v}{v_1} \times e_1$, where

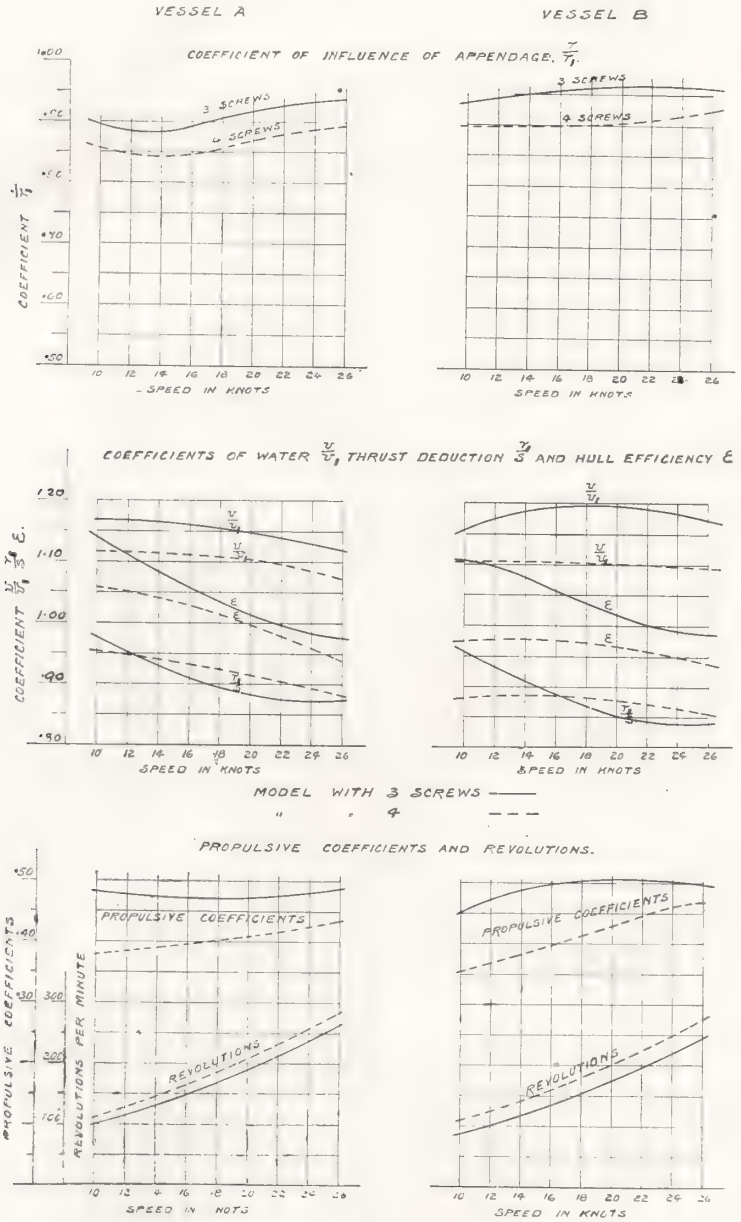


Fig. 2.

r = resistance of naked model

r_1 = resistance of model with appendages

$\frac{r_1}{s}$ = thrust deduction factor

$\frac{v}{v_1}$ = wake factor

e_1 = efficiency of the isolated screw.

The values of $\frac{r}{r_1}$ determined from the experiments are shown in Fig. 2. In both models this coefficient has considerably higher values for the three-propeller compared with the four-propeller arrangement, the difference at a speed of 25 knots amounting to about 8 per cent. in vessel A and 5 per cent. in vessel B. The values of the wake factor, thrust deduction factor, and hull efficiency $\epsilon = \frac{v}{v_1} \times \frac{r_1}{s}$ as determined by the experiments for different speeds are also shown in Fig. 2. It will be seen that the hull efficiency in the case of both models is considerably better with the triple screws than with quadruple screws. It was found that the gain in the wake factor obtained with the three-screw arrangement was effected at the central screw. The total propulsive efficiency E, obtained with the two arrangements, is shown in the lower diagram of Fig. 2. The total gain obtained at 25 knots by the use of triple screws is 12 per cent. for vessel A and 8 per cent. for vessel B.

TABLE I.

	Vessel A.		Vessel B.	
	3 Screws.	4 Screws.	3 Screws.	4 Screws.
Diameter D, metres	4·094	3·680	4·264	3·833
Pitch P, metres	3·875	3·360	4·037	3·50
$\frac{P}{D}$	·9465	·913	·9465	·913
Projected area, sq. metres	7·95	5·87	8·63	6·37

SCREW PROPELLER DESIGN.

By Captain C. W. DYSON, U.S.N.

(*American Society of Naval Engineers ; August*, 1911.)

THE method of screw propeller design developed by the author is based upon the analysis of trial trip results of actual ships. In his analysis the author employed the constants proposed by Barnaby based upon Froude's results, *viz.* :—

$$C_A = \frac{A V^3}{I.H.P.}$$

and $$C_R = \frac{R \times D}{V}$$

Assuming that, for propellers identical in all respects except that of developed area, the thrust that can be delivered will vary directly as the developed areas, Barnaby's constants can be represented graphically as a chart, and this has been done, as shown in Fig. 1.

While for designing purposes this chart is rough and ready, as it assumes the E.H.P. for the bare hull as equal to 50 per cent. of the I.H.P. required to propel the vessel, neglects appendage resistance entirely, and assumes a standard wake and standard thrust deduction of 10 per cent. each, it can be readily used to obtain factors from actual trial results of full-sized vessels and propellers, from which a more accurate method of design may be derived. The chart has been used in this way by the author to obtain a corrective factor which allows for the correct wake and thrust deduction in the case of the trials analyzed. To do this, he replaced V in the formulæ for C_A and C_R by $V = M \times S$, and then proceeded to ascertain from the chart the corresponding value of M, S being the actual speed of the ship in knots

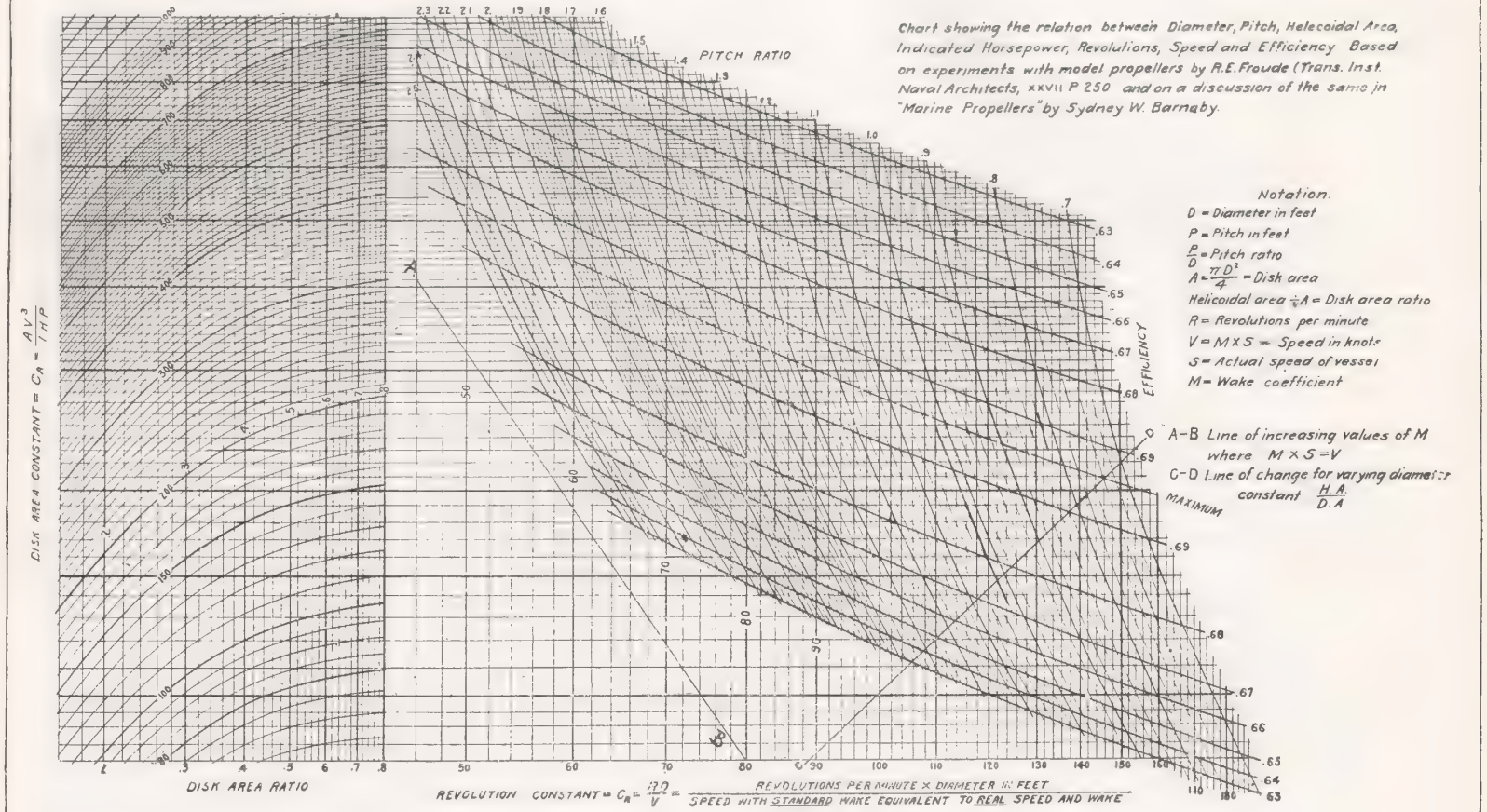


Chart showing the relation between Diameter, Pitch, Helecoidal Area, Indicated Horsepower, Revolutions, Speed and Efficiency Based on experiments with model propellers by R.E. Froude (Trans. Inst. Naval Architects, xxvii P 250 and on a discussion of the same in "Marine Propellers" by Sydney W. Barnaby.
PITCH RATIO
EFFICIENCY
MAXIMUM
DISK AREA CONSTANT = $C_A = \frac{AV^3}{I.H.P.}$
DISK AREA RATIO
REVOLUTION CONSTANT = $C_R = \frac{R.D}{V}$ = REVOLUTIONS PER MINUTE X DIAMETER IN FEET / SPEED WITH STANDARD WAKE EQUIVALENT TO REAL SPEED AND WAKE
Notation.
D = Diameter in feet
P = Pitch in feet.
$\frac{P}{D}$ = Pitch ratio
$A = \frac{\pi D^2}{4}$ = Disk area
Helicoidal area ÷ A = Disk area ratio
R = Revolutions per minute
V = M x S = Speed in knots
S = Actual speed of vessel
M = Wake coefficient
A-B Line of increasing values of M where M x S = V
C-D Line of change for varying diameter constant $\frac{H.A.}{D.A.}$

Fig. 1.—Chart of Design for Screw Propellers.

per hour. In the case of progressive runs values of M were calculated for each speed and plotted to a base of indicated thrusts in lb. per square inch of disc area of propeller. On this same base were erected the corresponding curves of apparent slip, of tip speeds of the propeller, and of the propulsive coefficients; while the block coefficient of fineness of the vessel, the projected area ratio of the screw $\frac{PA}{DA}$, and the shape of the projected area were noted on the same diagram. Having collected a large number of these preliminary curves from as many differently formed vessels as possible, the author was in a position to develop final charts of design.

and amount of blade surface of the propeller. From an examination of the performances of a number of propellers giving very good results, the author proposes the standard form shown in Fig. 2.

The final charts consist of :—

1.—A chart showing values of M for various values of $\frac{PA}{DA}$ plotted in relation to block coefficients (Fig. 3). The basis curves are those for $\frac{PA}{DA}$ ·32 and ·54, these being the values of $\frac{PA}{DA}$ in the case of the best screws analyzed.

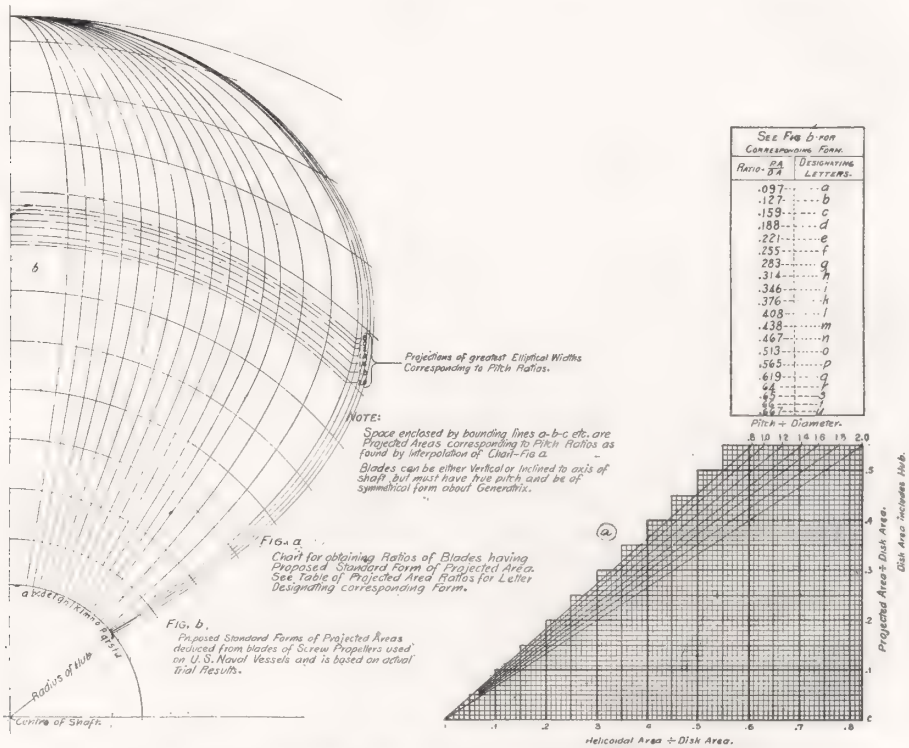

See Fig b for Corresponding Form.
Ratio PA/DA
Designating Letters.
.097 a
.127 b
.159 c
.188 d
.221 e
.255 f
.283 g
.314 h
.346 i
.376 k
.408 l
.438 m
.467 n
.513 o
.565 p
.619 q
.64 r
.65 s
.667 t
Projections of greatest Elliptical Widths Corresponding to Pitch Ratios.
NOTE:
Space enclosed by bounding lines a-b-c etc. are Projected Areas corresponding to Pitch Ratios as found by Interpolation of Chart-Fig a.
Blades can be either Vertical or Inclined to axis of shaft, but must have true pitch and be of symmetrical form about Generatrix.
Fig. a
Chart for obtaining Ratios of Blades having Proposed Standard Form of Projected Area. See Table of Projected Area Ratios for Letter Designating corresponding Form.
Fig. b.
Proposed Standard Forms of Projected Areas deduced from blades of Screw Propellers used on U.S. Naval Vessels and is based on actual Trial Results.
Radius of Hub
Centre of Shaft
Pitch ÷ Diameter.
Projected Area ÷ Disk Area.
Disk Area Includes Hub.
Helicoidal Area ÷ Disk Area.

Fig. 2.—Proposed Standard Form of Propeller Blades.

The performances of screw propellers are affected by the form of hull behind which they operate, by their location in regard to this hull, by their pitch, their diameter, their amount and distribution of blade surface, the condition of the blade surface, and by the fineness of the blade sections. Some of these factors can be kept nearly constant by the designer, and in general there remain only four variables to consider in the design, *viz.*, form of hull, pitch, diameter,

2.—A chart of apparent slips for different values of indicated thrust per square inch of disc area plotted in relation to block coefficient for the two values of $\frac{PA}{DA}$ ·32 and ·54 (Fig. 4).

3.—A chart of propulsive coefficients for various values of $\frac{PA}{DA}$ plotted in relation to indicated thrust per square inch of disc area, as shown in Fig. 5.

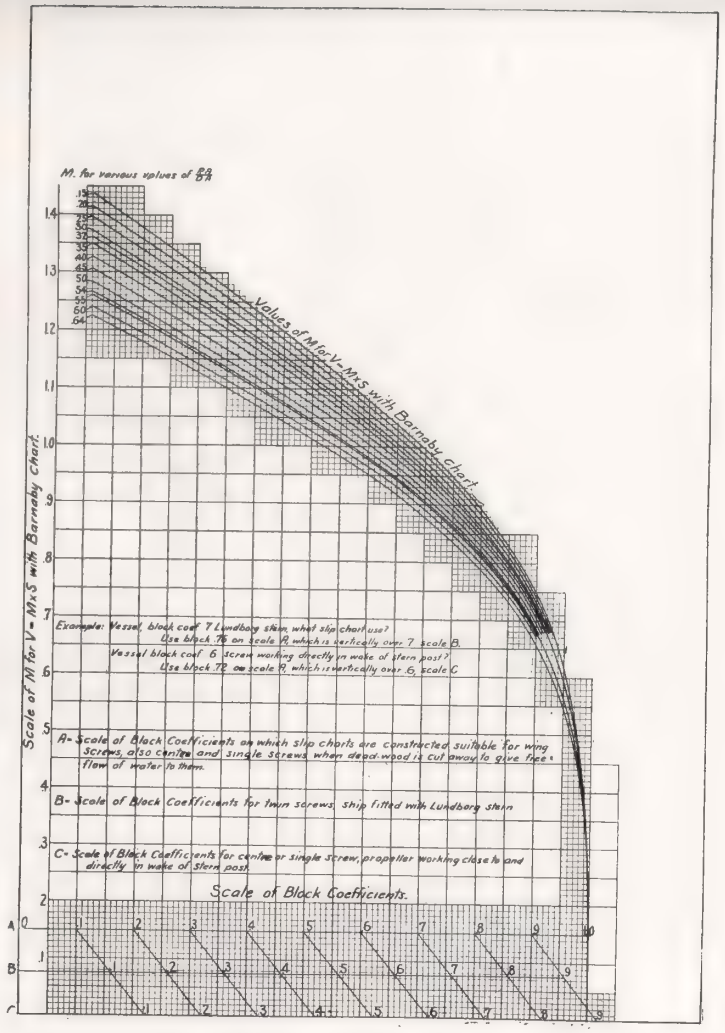

M. for various values of B/T
Values of M for V = MxS with Barnaby Chart.
Scale of M for V = MxS with Barnaby Chart.
Example: Vessel, block coef. 7 Lundberg stern, what ship chart use?
Use block 76 on scale A, which is vertically over 7 scale B.
Vessel block coef. 6 screw working directly in wake of stern post?
Use block 72 on scale A, which is vertically over 6, scale C.
A- Scale of Block Coefficients on which ship charts are constructed suitable for wing screws, o30 centre and single screws when dead wood is cut away to give free flow of water to them.
B- Scale of Block Coefficients for twin screws, ship fitted with Lundberg stern
C- Scale of Block Coefficients for centre or single screw, propeller working close to and directly in wake of stern post.
Scale of Block Coefficients.

Fig. 3.—Chart of *M* Values.

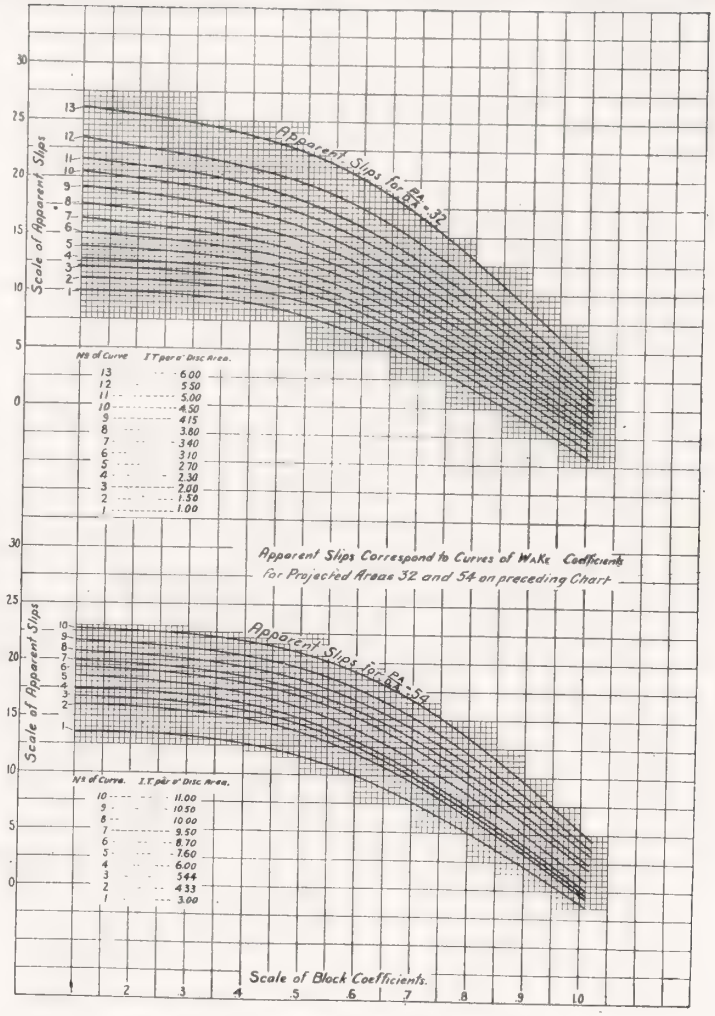


Fig. 4.—Chart of Apparent Slips.

4.—A chart of tip speeds for different values of $\frac{PA}{DA}$ plotted in relation to indicated thrust per square inch of disc area, as shown in Fig. 6.

5.—Charts of apparent slips, fifteen in number, corresponding to different values of the block coefficient from ·25 to ·9, and giving the apparent slip for varying values of $\frac{PA}{DA}$ plotted in relation to indicated thrust per square inch of disc area. These charts were constructed from the apparent slip

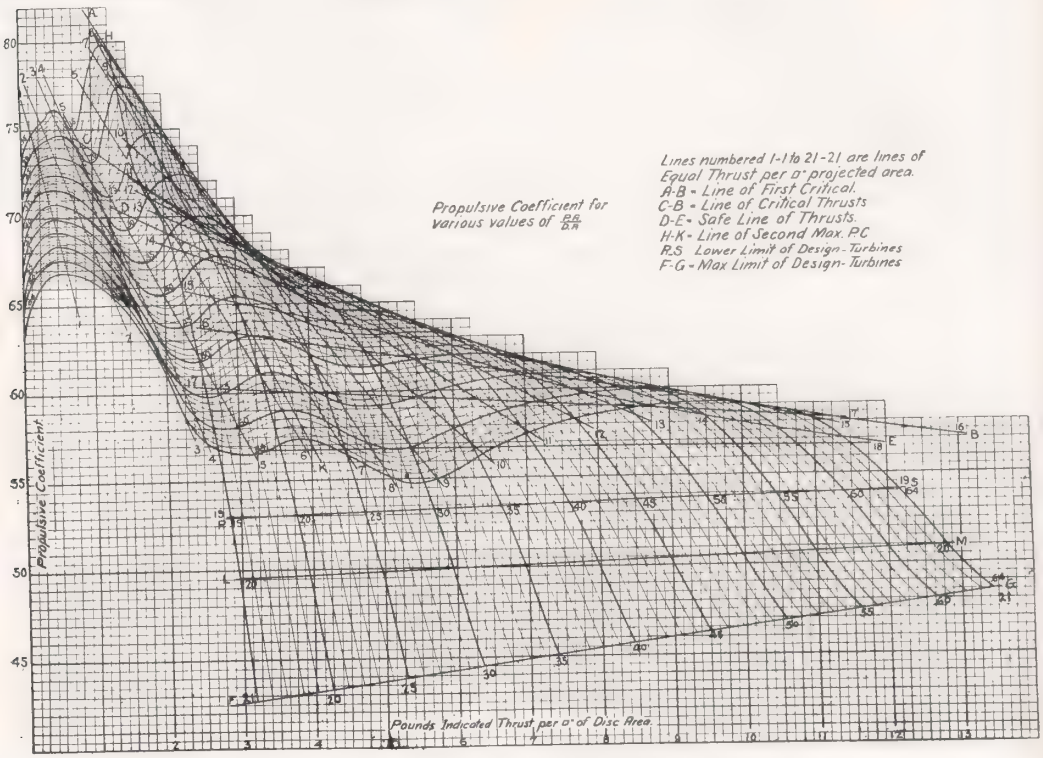

Lines numbered 1-1 to 21-21 are lines of Equal Thrust per □" projected area.
A-B - Line of First Critical.
C-B - Line of Critical Thrusts
D-E - Safe Line of Thrusts.
H-K - Line of Second Max. PC
R.S Lower Limit of Design - Turbines
F-G - Max Limit of Design - Turbines
Propulsive Coefficient for Various values of PA/DA
Propulsive Coefficient.
Pounds Indicated Thrust per □" of Disc Area.

Fig. 5.—Chart of Propulsive Coefficients.

curves given in Fig. 4. One of the charts, that for ·55 block coefficient, is given in Fig. 7.

The diagonal lines marked 1–1, 2–2, etc., up to 21–21, and shown on all the charts, are lines of equal indicated thrust per square inch of projected area from 1 lb. to 21 lb. The charts can be used without correction for three-bladed propellers placed well clear of the hull with a free flow of water to them. Screws working directly in the wake of and close to the sternpost require the correction for block coefficient indicated in Fig. 3. To design a four-bladed propeller from the charts, multiply the E.H.P. of the hull with all appendages by ·865, and use this result as the E.H.P. in the calculations. Multiply the projected area ratio of the screw thus obtained by $\frac{4}{3}$, and the result will be the necessary projected area ratio for the four-bladed screw.

The basic assumptions upon which the charts have been constructed are :—

1.—That for any fulness of hull, the projected area, pitch, and diameter of the propeller are dependent on each other, and must vary together.

2.—That with any particular projected area ratio, the same efficiency of propulsion can be realized, no matter what the fulness of hull, provided the pitch and diameter of the screw are so adjusted as to properly meet the conditions existing in the wake of the hull.

3.—That the velocity of the water through the propeller is not only dependent on the form of hull, but also on the propeller itself, and therefore on the projected area ratio of the propeller.

4.—That as the tip speed of the screw

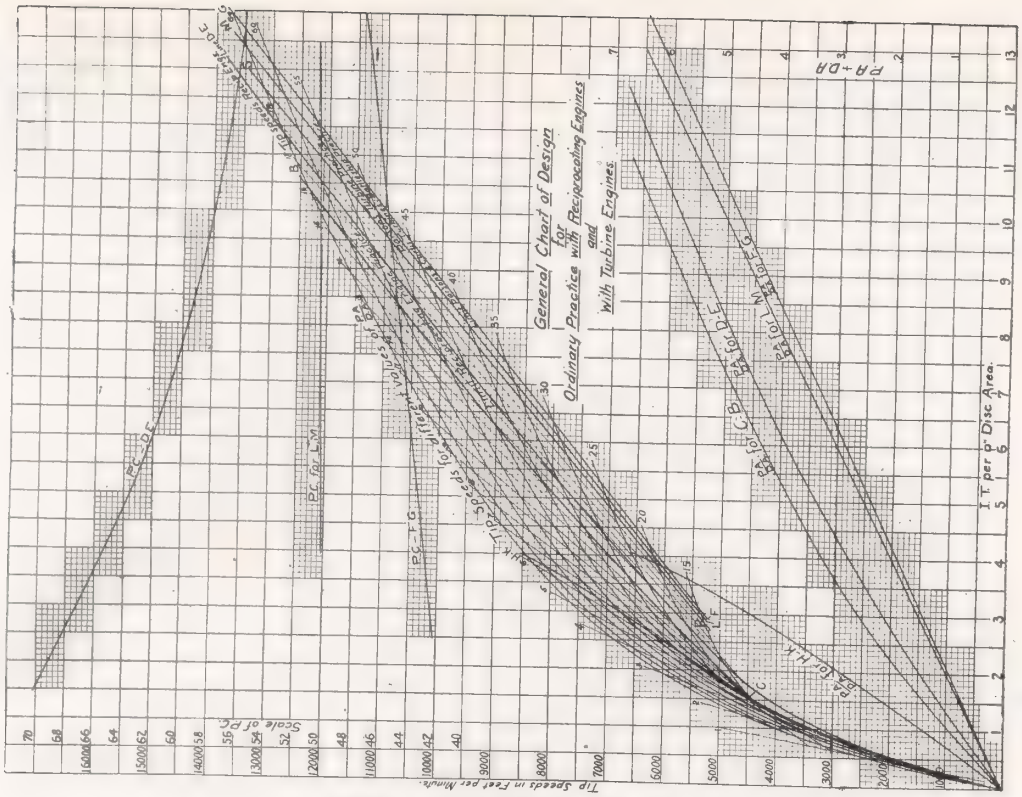


Fig. 6.—Chart of Tip Speeds.

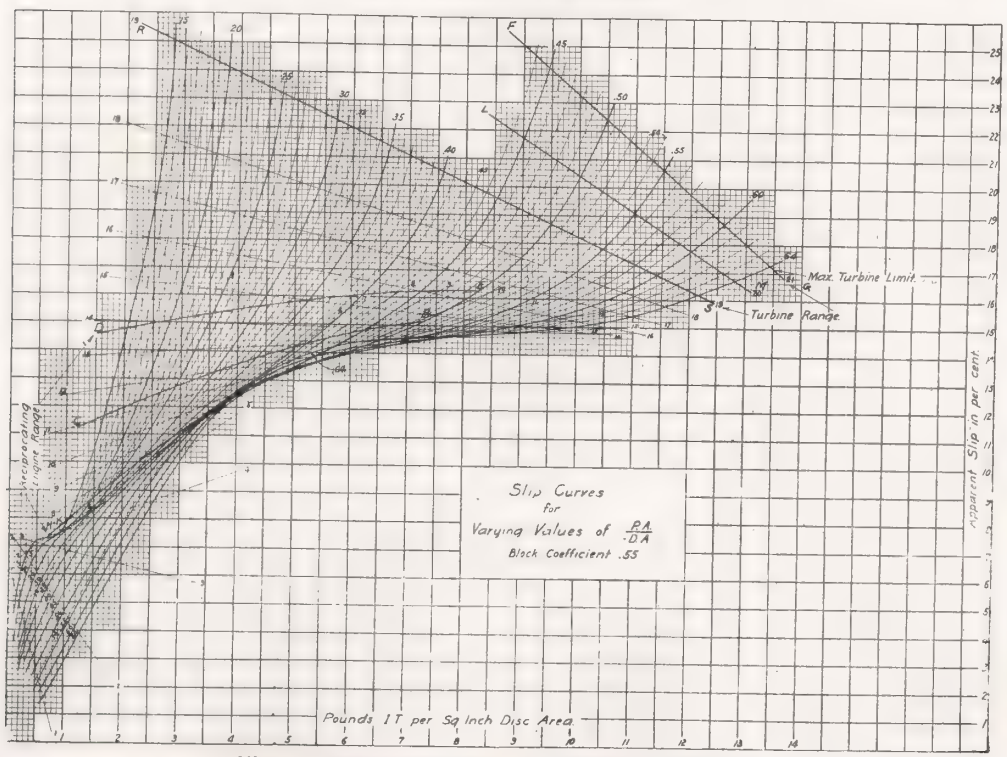


Fig. 7.—Slip Curves for ·55 Block Coefficient.

varies with its pitch and its diameter, and as these vary with the projected area ratio, the tip speed must vary with this ratio in a similar manner.

5.—That as the apparent slip for any fulness of hull depends on the pitch and the diameter of the propeller, and as these vary with the projected area ratio, so will the apparent slip vary with it in like manner.

6.—That shaft horse-power is equal to 92 per cent. of indicated horse-power.

Numerous examples are given by the author showing how to apply the charts, and as a further guide an extensive table of the hull data of actual ships, with the corresponding propellers and performances, is appended.

The model-tank curves of E.H.P. as usually furnished to the designer are for the bare hull only, and require the addition of the E.H.P. necessary to overcome the resistance of the appendages before the problem of the propeller can be approached. To provide a means of estimating the effect of the appendages, the author submits the following table :—

A = Large appendages, not well placed; axes of strut arms fore and aft; two shafts; reciprocating engines.
B = Large appendages, carefully placed; axes of strut arms parallel to stream lines; two shafts; reciprocating engines.
C = Moderate-sized struts; appendages carefully placed as in *B*; two shafts; turbine engines.
D = Small-sized struts; appendages carefully placed as in *B* and *C*; three or four shafts; turbine engines.
E = Single-screw ships; appendages carefully placed.

Reference Letter.	Per cent. E.H.P., bare Hull.	Per cent. E.H.P., Struts, Shafts, Bosses.	Per cent. E.H.P., Bilge Keels.	Per cent. E.H.P., Docking Keels.	Per cent. E.H.P., Rudder.	Per cent. E.H.P., large Scoops.	Total per cent. E.H.P., of bare Hull.
A	100	9·4	4·25	5·5	0·85	2·0	122
B	100	8·0	3·05	5·1	0·85	2·0	119
C	100	5·0	3·05	5·1	0·85	2·0	116
D	100	3·0	3·05	5·1	0·85	2·0	114
E	100	0·0	3·05	5·1	0·85	2·0	111

SCREW PROPELLER DESIGN: THE INFLUENCE OF THE RATIO BETWEEN GREATEST IMMERSED BEAM AND LENGTH ON LOAD WATER-LINE ON BLOCK COEFFICIENT AND ON THE CHOICE OF SLIP CHART TO USE FOR ANY GIVEN VESSEL.

By Captain C. W. Dyson, U.S.N.

(*American Society of Naval Engineers; February*, 1912.)

THE author presents a corrective chart (Fig. 1) for block coefficients for guidance in determining the proper slip chart to use when following the method of propeller design developed in his earlier paper. The necessity for this correction arises from the fact that the same form of after body, and therefore flow of water to the propeller, may be associated with different values of block coefficient in ships of different lengths, in which the change of length occurs in the parallel middle body.

The block coefficients for a series of vessels derived from a parent form by adding midship body, if plotted in relation to values of $\frac{B}{L}$, will be found to lie in a straight line passing through the axis of the abscissæ at a block coefficient equal to 1·0, and through the point on the chart corresponding to the block coefficient and the value $\frac{B}{L}$ of the standard ship. This fact having been established by trial, all that is necessary in constructing the corrective chart for block

coefficients is to establish the line of parent (standard) $\frac{B}{L}$ and of parent (standard) coefficient of immersed midship section. Then all vessels of approximately equal fineness of fore and after bodies may be considered to lie on a line drawn through the parent ship point and the point marked *D* on the chart. The coefficient of midship section is given directly above the point corresponding to the parent ship $\frac{B}{L}$. Thus, on the chart, *A* of nominal block coefficient = ·78, and *E* of nominal block coefficient = ·61, are taken so that both lie on the line *OE*, intersecting the standard line of $\frac{B}{L}$ at *B*, corresponding to a block coefficient of ·667. The coefficient of immersed midship section corresponding to the latter block coefficient is ·98. The apparent slip for the design of the propeller would be obtained by interpolating between slip charts for block coefficient ·65 and ·70.

Should the immersed midship coefficient be considerably greater or less than the standard value on the chart, then the end lines would be finer or fuller than the standard form respectively, and an allowance must be made on this account, according to the experience of the designer. If, however, the ship is normal in her lines, with no wide departure in coefficient of immersed midship section from the standard as given by the curve, the chart (Fig. 1) may be used with confidence.

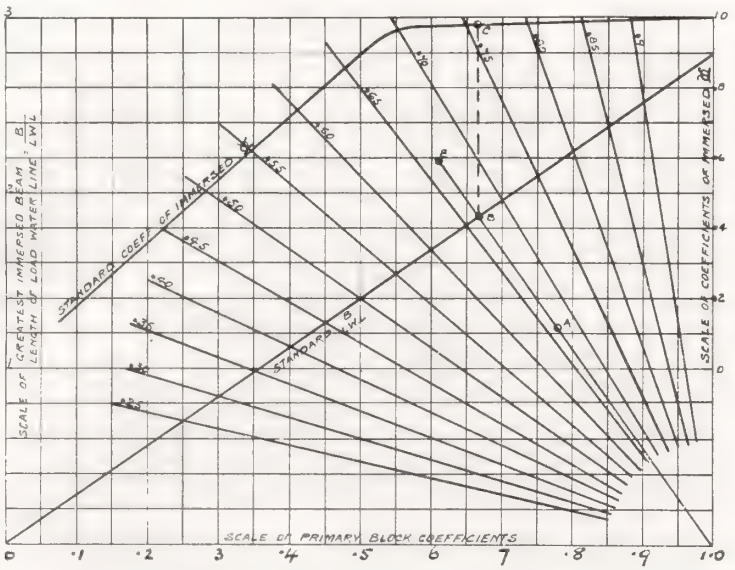

SCALE OF GREATEST IMMERSED BEAM / LENGTH OF LOAD WATER LINE = B/LWL
STANDARD COEFF. OF IMMERSED
STANDARD B/LWL
SCALE OF COEFFICIENTS OF IMMERSED
SCALE OF PRIMARY BLOCK COEFFICIENTS
0 ·1 ·2 3 ·4 ·5 6 7 ·8 ·9 1·0

Fig. 1.—Corrective Chart for Block Coefficients.

PRACTICAL EXPERIENCE WITH COUNTER-PROPELLERS.

By Dr. R. Wagner.

(*Schiffbautechnische Gesellschaft; November*, 1911.)

THE device to which Dr. Wagner has given the name "gegen propeller" (counter propeller) consists of an arrangement of fixed blades, which are placed behind the ship's propeller. Its function is to convert into useful work the energy which is otherwise wasted by the revolving propeller in imparting a spiral instead of a purely axial motion to the particles of water in the propeller race. The action of the counter propeller will be understood from Fig. 1. The initial motion of the water acted upon by a propeller blade takes place in a direction inclined to that in which the propeller moves forward. If *ab* is taken to represent the velocity of the water in

this inclined direction, then *ac* represents the component of this velocity, to which the effective thrust of the propeller is proportional. If, however, a fixed blade of correct form is introduced, as shown, the motion of the water will be changed to an axial direction theoretically, without any

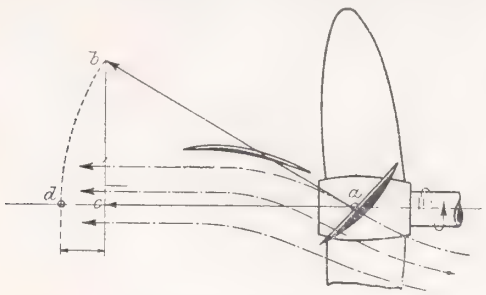


Fig. 1.—Action of Counter-Propeller.

reduction in the velocity, *ad* with the counter propeller being equal to *ab* without, the result being a gain in effective thrust proportional to the difference between *ac* and *ad*.

The highest efficiency attained with present-day propellers is from 65 to 75 per cent., or considerably less than that of modern high-class water turbines. By adding the counter propeller the author claims that the efficiency can be increased to 85–87 per cent., and probably even to 90 per cent. when further experience has been gained. Counter propellers have been fitted with success to a number of screw yachts, three small steamers, and to Torpedo-Boat Destroyer "V 185" belonging to the Imperial German Navy; while their adoption for numerous other vessels, including large cargo steamers, is now under consideration. A photograph of the stern of "V 185" showing the counter propellers is reproduced in Fig. 2. The vessel is propelled by twin screws driven by turbines, the total shaft horse-power being about 16,000. Trials run with this vessel, both with and without counter propellers, showed that a considerable reduction in the power required for a given speed was effected by their use. The results of the trials are given in Fig. 3. For the full speed of 32·6 knots, the shaft horse-power without counter propellers was 17,400, and with counter propellers, 15,600, the reduction in the power required for this speed due to their presence being, therefore, about 10 per cent. On the basis of the successful results achieved with counter propellers in this and other vessels and the theoretical investigations he has made, the author considers that in the case of twin and multiple-screw ships a saving of at least

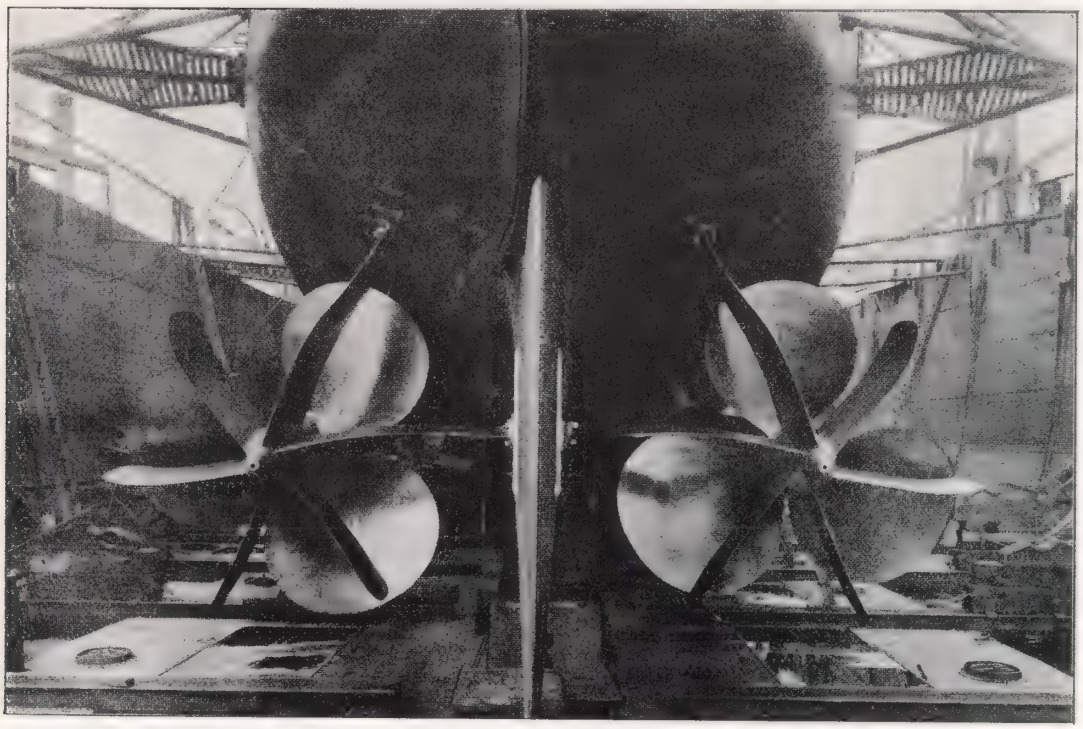


Fig. 2.—Torpedo-boat Destroyer "V 185": View of Stern showing Counter-Propellers.

10 to 15 per cent., and in the case of single-screw ships a saving of at least 8 to 12 per cent., in high speed vessels when running at full speed. In the case of canal boats the counter propeller

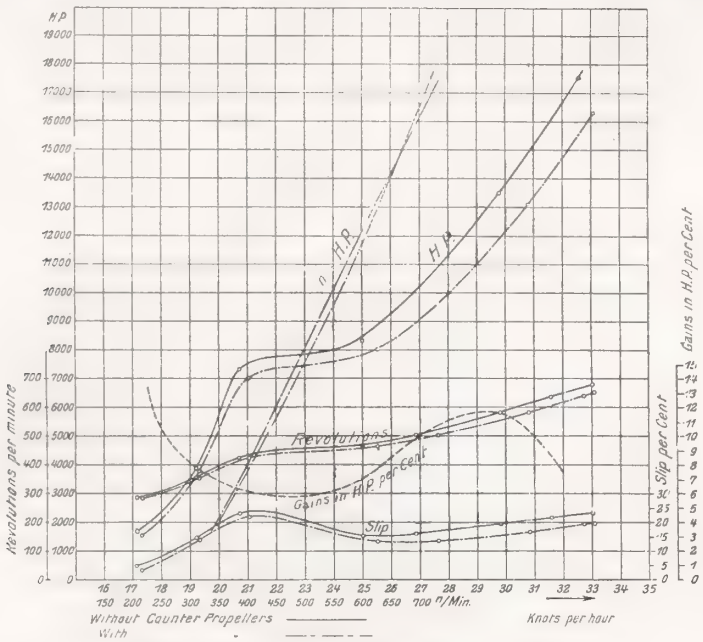


Fig. 3.—Results of Trials of "V 185."

power can be effected by their adoption. A further advantage of the counter propeller is that it reduces vibration and lessens sinkage of the stern tends to reduce the erosion of the bottom and banks of the canal. Counter propellers can also be adopted with great advantage for airships.

AUTOMATIC RECORD OF PROPELLER ACTION IN AN ELECTRICALLY-PROPELLED VESSEL.

By Mr. W. L. R. Emmet.

(*American Society of Naval Architects and Marine Engineers; November*, 1911.)

In studying the problems connected with electric ship propulsion, the author has encountered some contradictions and differences of opinion concerning the conditions necessary for satisfactory reversal; and the lack of definite information which has been discovered led to the making of the experiments on the fire-boat *Graeme Stewart* which are recorded in the paper. This vessel has twin screws, each driven by a motor, to which current is supplied by the turbine-driven direct-current generators installed for working the fire pumps. The fields of these motors are separately energized from a constant-potential exciting generator, and speed changes are accomplished by changing or reversing the field excitation of the generators. A record of current variation during each reversal test was obtained by means of a specially designed recording ammeter, the apparatus being so arranged that the current variations, propeller revolutions, and equal time intervals were marked side by side on a strip of paper. In addition to these records a propeller log was rigged on an outrigger near the bow of the boat, and connected through an electric circuit in such a manner that a mark was made upon the paper strip every time the propeller made a revolution. The log was calibrated by repeated runs over established distances, and from the results of this calibration the speed of the vessel could be accurately determined from the marks on the paper down to a speed of about 150ft. per minute. From the records Fig. 1 has been prepared, showing the relation of speed to power, slip, and

average revolutions per minute of the two propellers, and Fig. 2, showing the relation between

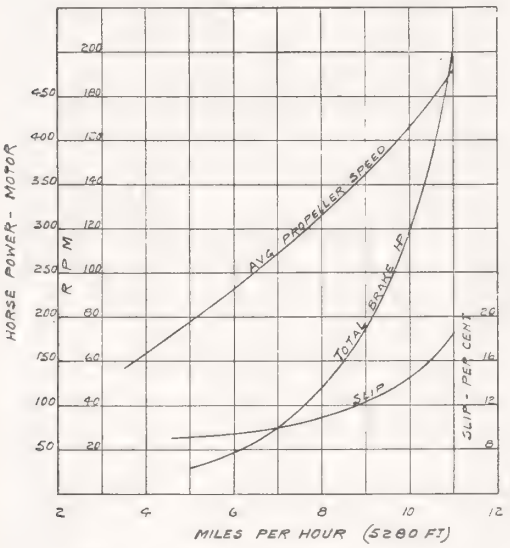


Fig. 1.—Trial Records of the "Graeme Stewart."

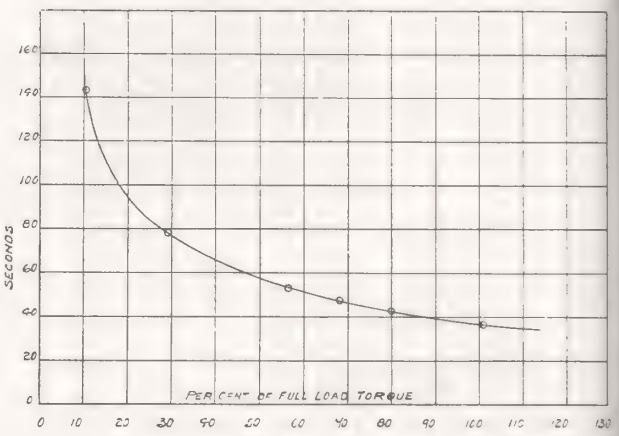


Fig. 2.—Relation between Torque and Time required to stop the Vessel.

the torque and the time required to stop the vessel. In considering the reversal conditions shown by these tests the propeller characteristics, as indicated by the slip curve, should be borne in mind. The maximum speed which the vessel is capable of making is 11 miles per hour, and at this speed the slip is nearly 19 per cent., while at 5 miles per hour it is only 9 per cent. The increase of slip at higher speeds indicates that the propeller is of insufficient size, a deficiency which would tend to diminish the effect of the higher torque values in the reversal experiments.

NEW EXPERIMENTAL RESEARCH ON THE RESISTANCE OF AIR AND AVIATION.

By Monsieur G. Eiffel.

(*Société des Ingenieurs Civils de France; May*, 1912.)

THE present paper is an extension of the paper communicated by M. Eiffel in December, 1911, and deals with the latest results obtained at the author's Champ-de-Mars laboratory and his new laboratory at Auteuil mainly in connection with the study of propellers. In order to be of real value, the experiments must be carried out in a relative wind, the screw being either in forward motion in still air or stationary in a uniform current of air. The latter method was adopted at Champ-de-Mars. The model and dynamo controlling it were suspended in a current of air in such a way that the turning moment was measured by the inclination of the apparatus and the thrust by an aerodynamic balance.

According to the usual hypothesis, the air resistance of the propeller blade is proportional to the area and to the square of the speed, and it is assumed that the blades retain their shape. The elements of the screw can then be represented by a single curve under all circumstances, and this curve applies to all screws geometrically similar. It suffices to take for abscissæ the ratios of the speed of advance to the circumferential velocity of the screw or $\frac{V}{nD}$ and for ordinates the values $\frac{F}{n^2D^4}$, $\frac{G}{n^2D^5}$, etc., F being the thrust and G the turning moment, etc.

Experience has shown, however, that in reality this single curve is replaced by curves varying with the speed. At high speeds the resistance ceases to vary with the square of the speed, while deformations also occur in the propeller blades. Nevertheless from the study of a model propeller

MONSIEUR G. EIFFEL,
The eminent French Scientist and Engineer.

G. Eiffel

it is possible to determine the elements of screws geometrically similar. It suffices to make the experiments with the same velocity of wind and at speeds of rotation inversely proportional to the diameters of the screw and its model (for example 3,600 revolutions for a model one-third the size of a screw turning at 1,200 revolutions). Under these conditions, provided the model and screw are of the same material, they are in effect geometrically similar, as the relative deformations and speeds are the same.

To test the above theory, M. Dorand tried a full-sized propeller of the Dizewiecki type on his dynamometer car at Chalais, and M. Eiffel a model of this propeller on a scale of one-third full size. It was found that within the values of speeds occurring in practice the curves of the results were nearly coincident at the same values for nD. In order to give the model propeller the high rate of revolution required, the new laboratory has been provided with a powerful experimental apparatus capable of maintaining a speed of 4,000 revolutions.

The author then gives the results of his latest experiments upon the resistance of flat and curved planes of different areas and lengths. He also deals with the influence of the thickness of the plane and with the effects produced when the angle of the plane is varied from the centre to the sides.

In conclusion, the author describes his new laboratory at Auteuil, which is in general similarly arranged to that at Champ-de-Mars, but embodies several improvements suggested by experience with the earlier apparatus. The Champ-de-Mars laboratory has furnished from 4,000 to 5,000 experimental results in connection with aerodynamics, and M. Eiffel hopes that the Auteuil laboratory will permit him to still further accumulate results of scientific and industrial value.

AERIAL FLIGHT.

By Mr. H. R. A. Mallock, F.R.S.

(*The James Forrest Lecture delivered at the Institution of Civil Engineers; April*, 1912.)

THE portion of Mr. Mallock's lecture dealing with the motion of fluids round inclined planes is of great interest to naval architects as well as to those concerned with aeronautics. Mr. Mallock first considered the theoretical side of this part of his subject.

In the case of a thin flat plate placed edgewise in a wind, if there is no surface friction between the fluid and the plane, the latter offers no resistance, the pressure on both sides is the same, and the flow of the air-stream is unaltered by the presence of the plate. If, however, the plane is inclined to the current, the mathematical solutions present two possible forms of flow. One of these would indicate that the plane experienced no resistance, but that the pressure was so distributed over the two surfaces as to form a couple tending to turn the plate broadside to the stream (Fig. 1). The other would indicate a resistance, the pressure on the up-stream surface being raised, while on the down-stream surface the pressure remains unaltered, equal, that is, to the general pressure in the air at a distance. This latter form of flow is shown in Fig. 2, and it will be seen that behind the plane there is a body of still air, separated from the current by a surface of discontinuity.

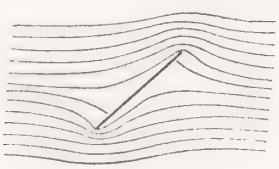


Fig. 1.

Neither of these theoretical flows represents what actually takes place, but the second is sufficiently close to nature to give a rough measure of the real resistance when the inclination of the plane to the direction of the current is a moderate angle, say not less than 8° or 10°.

From the theory of flow shown in Fig. 2, it appears that the average normal pressure on the unit area of the plane is proportional to

$$v^2 \frac{\pi \sin \alpha}{4 + \pi \sin \alpha},$$

where v is the velocity and α the angle which the plane makes with the stream. When this angle is small, this expression may be replaced by $v^2 \frac{\pi}{4} \sin \alpha$, and the actual normal force is $\rho v^2 \frac{\pi}{4} \sin \alpha$ per unit area of plane, ρ being the density of the fluid. Hence the components parallel and perpendicular to the flow are

$$\rho v^2 \frac{\pi}{4} \sin^2 \alpha \text{ and } \rho v^2 \frac{\pi}{4} \sin \alpha \cos \alpha.$$

The first of these is the resistance, and the second the lateral or lifting force, and their ratio is *cot.* α.

The general relations connecting the normal force, lateral force, resistance and power expended when a stream of fluid flows obliquely past a long narrow plane (the long dimension being across the stream) are given in the following formulæ :—

Let v denote the velocity of the stream.
α ,, ,, angle between the stream and plane.
S ,, ,, area of the plane.
N ,, ,, normal force on the plane.
R ,, ,, resistance.
L ,, ,, lateral force.
H ,, ,, power required in ft.-lb. per sec.
ρ ,, ,, density of the fluid.

Then on the theory of flow shown in Fig. 2 :—

$$N = \tfrac{1}{2}\,\rho\, v^2 \frac{2\,\pi \sin \alpha}{4 + \pi \sin \alpha} \qquad (1)$$

$$R = \tfrac{1}{2}\,\rho\, v^2 \frac{2\,\pi \sin^2 \alpha}{4 + \pi \sin \alpha} \qquad (2)$$

$$L = \tfrac{1}{2}\,\rho\, v^2 \frac{\pi \sin \alpha \cos \alpha}{4 + \pi \sin \alpha} \qquad (3)$$

When the angle is small enough for α to be sensibly equal to sin α, and cos α is nearly unity, if we put $\frac{1}{2}\,\rho \frac{2\,\pi}{4} = A$.

$$L = N = AS\, v^2\, \alpha \qquad (4)$$

$$R = AS\, v^2\, \alpha^2 \qquad (5)$$

$$H = vR = AS\, v^3\, \alpha^2 \qquad (6)$$

$$\frac{L}{R} = \frac{1}{\alpha} \qquad (7)$$

In Fig. 3 the ordinates of the dotted curves are the L and R values for the equations just

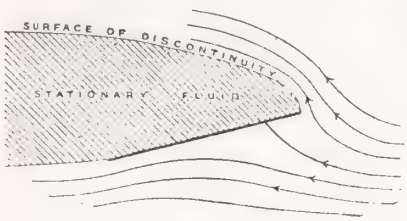


Fig. 2.

given, and they refer to a plane whose breadth parallel to the stream is small compared with its length. The units in which these curves are plotted are such that 1 corresponds to the weight of a column fair of unit section whose height is the dynamic head required for the velocity. The ratio $\frac{L}{R}$, which measures the proportion of the lateral force to the resistance (or, in the case of flying, the lifting force to the propulsive effort), is simply a curve of cotangents M. The full curves in Fig. 3 are the result of experiments made with planes moving in air or in water, as for most purposes the results obtained from water may be applied to air and *vice versâ*, provided the relative densities of the two fluids are taken into account.

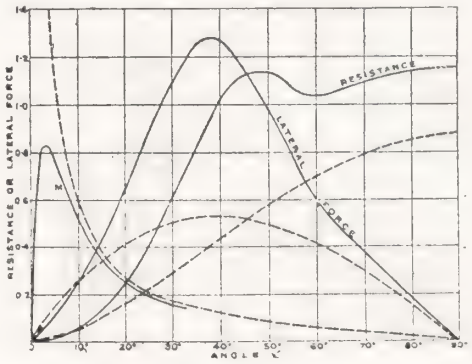


Fig. 3.

Three points in particular stand out clearly in the experimental curves, *viz.* :—

1.—That the actual resistance and lateral force are greater than the theoretical values.

2.—That their ratio ceases to be a curve of cotangents, and that it has a large and distinct maximum when α is somewhere about 5°.

3.—That both the resistance and lateral force are very uncertain when α is within 10° or 15° on either side of 45°.

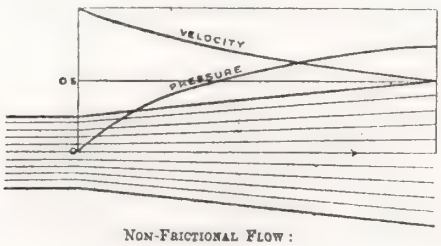
The lecturer then proceeded to consider the causes which produce the very marked difference between the theoretical and experimental curves given in Fig. 3.

The fluid with which mathematicians deal, and which is supposed to surround the plane in Fig. 2. is an ideal fluid without viscosity (that is, opposes no resistance to shear) and in contact with a solid it experiences no frictional retardation. In such a fluid pressure and velocity are connected by an invariable law, the sum of the potential and kinetic energies of any portion of the fluid remaining constant for all time.

This law, together with the necessary condition of continuity, which, for an imcompressible fluid, merely implies that the volume remains constant, no matter what shape it takes, constitutes the foundation of all the propositions regarding the stream lines of a perfect fluid which have hitherto been worked out, and for such a fluid the stream lines indicated in Fig. 2 are an exact solution of the problem.

Real fluids differ from the perfect fluid in having

both viscosity and surface friction. They require that work should be done if distortion is going on, and they adhere to the surfaces of solids immersed in them. Although for fluids such as water and air the direct effects of these factors are small, indirectly they have immense influence. The

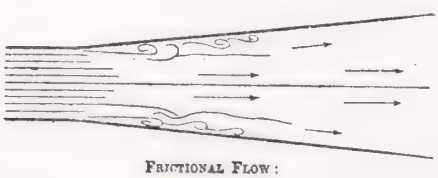


Non-Frictional Flow:

Fig. 4.

indirect action depends on the fact that when a stream is retarded by friction the velocity is reduced although the pressure remains unchanged, and thus the fundamental relation which connects velocity and pressure in a perfect fluid is violated. The effect of friction and viscosity is much more conspicuous when the stream is divergent (so that in the absence of friction the velocities and pressures, although constant across each section, change from one section to another, but keep the total energy of the flow across each section the same) than when the stream concerned is of constant section and is neither accelerating nor retarding.

In Fig. 4 the fluid is supposed to be flowing outwards in a wedge-shaped channel, and the stream-line pressure and velocity at each section for a perfect fluid are shown by the curves. This relation is the only one which will allow a portion of fluid at ρ_1 v_1 to overcome the resistance due to the pressure gradient and arrive at ρ_2 with the velocity v_2. In a frictional fluid these relations hold fairly well for the central streams, which are little influenced by the sides; but the side streams, which have had their velocity reduced by friction independently of the stream-line



Frictional Flow:

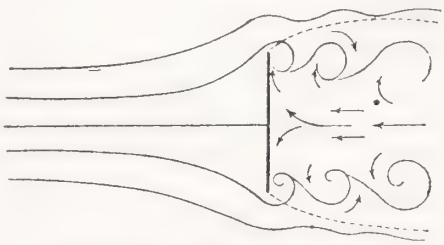
Fig. 5.

reduction, find themselves opposed by a pressure which is sufficient to reverse the direction of their flow, and these reversed streams by usurping part of the channel alter the character of the flow altogether; the result being that, instead of an outward flow over the whole section of the channel, a rapid current traverses the central part with well-marked eddies on either side (Fig. 5).

The exact nature and form of these eddies are not at present amenable to mathematical treatment, but their general character and the fact that they are due to the reversal of streams whose potential has been degraded by friction were pointed out by the late Mr. W. Froude before the British Association meeting at Bristol in 1875.

Turning again to Fig. 2, which shows the discontinuous flow of a perfect fluid past a plane, and comparing it with Fig. 6, the actual flow of water or air in the same circumstances, some of the reasons for the difference between the forces calculated and found will be seen.

On the up-stream side of the plane friction does little to modify the conditions except in the neighbourhood of the edges; but down-stream, instead of a pond of still fluid, there exists a complex wake consisting of a central current moving forward towards the plane, bordered by a series of eddies whose origin is of the same



Frictional Flow: Stream Normal to Plate.

Fig. 6.

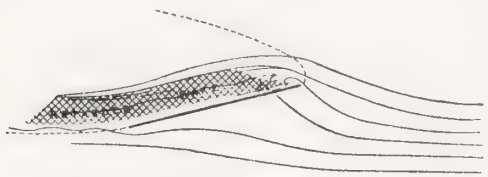
nature as those just referred to in the expanding channel, *viz.*, to degradation of the streams passing round the edges of the plane which, having insufficient velocity to follow the stream-line path of Fig. 2, are deflected inwards and become involved with the reversed central stream, about half the fluid in each eddy being supplied from up-stream and half from the wake.

The eddies are formed periodically, growing to a certain size, and then, breaking away from their place of birth, they form part of the train which borders the wake current. The wake current itself is due to the constant removal of fluid in this way from the back of the plane; and the fact that the outflow from the back has its maximum velocity close to the edge, where the composite eddy is being formed, shows that the pressure on the back of the plane is lower at the edges than in the centre. Hence it could be

stated with certainty, even without any experiment, that the total resistance of a plane must be greater than $\rho v^2 \frac{\pi \sin \alpha}{4 + \pi \sin \alpha}$, which assumes that the pressure over the rear surface is uniform, and equal to the general pressure at a distance.

Experiment, however, is required to determine the actual resistance, and when the plane is broadside to the stream this is found to be about half as much again as the head resistance alone, or about 20 or 25 per cent. greater than the dynamic head $\times$ the area of the plane.

When the angle α is small, as it always is in flight, the character of the wake takes the form shown in Fig. 7. Here the wake stream is only



FRICTIONAL FLOW: STREAM OBLIQUE TO PLATE.

Fig. 7.

recognisable as a reversed current quite close to the plane, and the small eddies as fast as they are formed are so rapidly degraded that after travelling a short distance they are merely recognisable as slight variations in the direction of the general current. The abstraction of wake water by eddy-making continues, however, even for very small values of α, and has the effect of deflecting the upper boundary of the wake as shown.

This short account gives a general explanation of the observed difference between results calculated for the discontinuous flow of a perfect fluid and those actually found by experiments in air and water, and if the nature of the flow over the back surface were accurately known, the value of α for the maximum of L/R could be predicted. Even in the absence of this knowledge, the assumption that surface friction varies as v^2, and acts only on the up-stream side, leads to a value of α that is not far removed from truth.

Let AB, Fig. 8, be the plane making a small angle α with the stream, and let L and R be the lateral force and resistance which would be experienced if there were no friction.

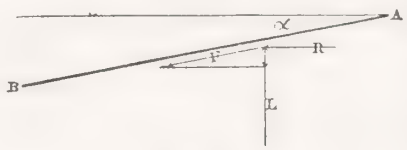


Fig. 8.

If L' and R' are the same quantities, taking friction into account, and putting $F v^2$ as the frictional force parallel to AB, we have $L' = L - F\alpha$, and $R' = R + F v^2$, and since $L = R_n \alpha$, and $R = R_n \alpha^2$, R_n being the normal resistance $A v^2$:

$L' = L - F\alpha = \alpha v^2 (A_n - F)$, and $R' = R + F v^2 = v^2 (A \alpha^2 + F)$; hence $L'/R' = \alpha (A - F) / (\alpha^2 A + F)$ and this is a maximum when $\alpha = \sqrt{F/R}$.

Lanchester's experiments make $F/R = 0{\cdot}0075$,
Zahn's ,, ,, ,, $F/R = 0{\cdot}0037$,

which correspond to $\alpha = 6{\cdot}5°$ or $3{\cdot}5°$ respectively.

The actual value found from direct experiments on L and R lies between these two, and although $6{\cdot}5°$ is nearer the truth than $3{\cdot}5°$, this does not imply that $0{\cdot}0075$ is the more nearly correct value of F/R, for the complete theory must take into consideration the action of the streams on both sides of the plane.

The work done in eddy-making, and therefore the total resistance, varies with the stage of growth of the eddy, and the resistance of a plane in a real fluid is not a constant, but a fluctuating quantity, the period of fluctuation being the time occupied in the formation of each eddy.

The whole subject of eddy production—real eddies, it must be remembered, are very different from the closed-circuit vortices of mathematics—presents a wide field for useful investigation, in which very little has as yet been done.

The Vibration of Ships.

TORSIONAL VIBRATIONS OF ELASTIC SHAFTS OF ANY CROSS SECTION AND MASS DISTRIBUTION, AND THEIR APPLICATION TO THE VIBRATION OF SHIPS.

By Professor Dr. L. Gümbel.

(*Institution of Naval Architects ; March*, 1912.)

IN a paper presented to the Institution of Naval Architects in 1902, Dr. Gümbel dealt with certain problems relating to the torsional vibrations of shafts. In the present paper the author extends his method of treatment to shafts having a straight axis and of any cross section and mass distribution. Whatever are the moments of inertia of the various cross sections of the shaft, the latter may be replaced by a shaft of uniform polar moment of inertia, the length being such that the distortions of the two shafts at all points which correspond to one another shall be the same under the influence of the same forces. Further, whatever be the distribution of the masses connected with this shaft, each mass may be replaced by a single mass at a distance r from the axis of gravity of the shaft sections.

amplitudes and frequencies of the free vibrations of the equivalent shaft. At the moment of maximum deformation of this shaft, there will be acting at the point of application of the masses m at radius r the forces $m\omega^2 r\alpha$, where ω is the angular velocity of the vibration and $r\alpha$ the amplitude. No other forces being at work, the sum of all these forces $m\omega^2 r\alpha$ must equal zero. Further, the author shows that the inclination of the two end tangents $\left(\frac{d\alpha}{dL}\right)$ of the curve of amplitudes is zero.

To construct the curve of amplitudes, make the polar distance in a force diagram equal to $\frac{I\,G}{r^2}$ where G is the modulus of rigidity, and plot the respective forces $m\omega^2 r\alpha$ successively, both in magnitude and direction. Assuming ω to be

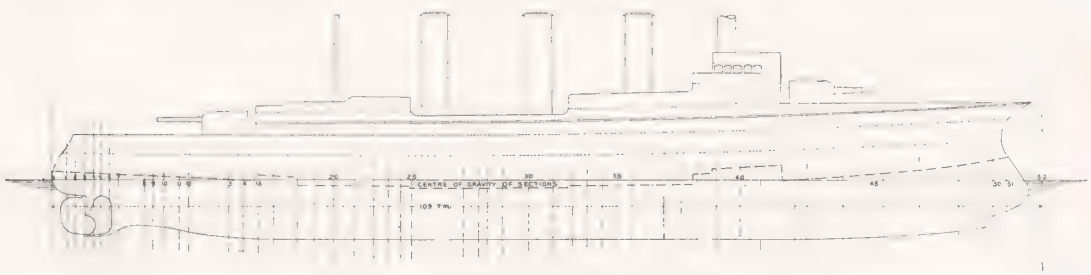


Fig. 1.—Small German Cruiser "Vineta."

The manner in which the equivalent shaft is obtained is illustrated by Figs. 1 and 2, which relate to the German cruiser *Vineta*. In Fig. 2 the moments of inertia of the masses per unit of length of the ship, referred to the axis of gravity of the sections, are distributed over the corresponding lengths of the equivalent shaft to represent the masses acting upon it. Taking a value $I = 10m^4$ for the uniform moment of inertia of the equivalent shaft, the total length of the ship, 109·7m., is replaced by the length of the equivalent shaft 88·54m.

The author then proceeds to determine the

known, the corresponding link polygon will yield the amplitudes of the equivalent shaft. As ω is in general not known, it is necessary to try several values and find by interpolation the curve of amplitude which will fulfil the condition $\frac{d\alpha}{dL} = 0$. In Fig. 2 the curves of amplitude have been obtained by selecting a polar distance $\frac{I\,G\,g}{\omega^2 r^2}$ and plotting the quantities $W r \alpha$ instead of the forces, where g is the acceleration and $W = mg$. It will be seen from Fig. 2 that two trial curves were drawn and the required curve found,

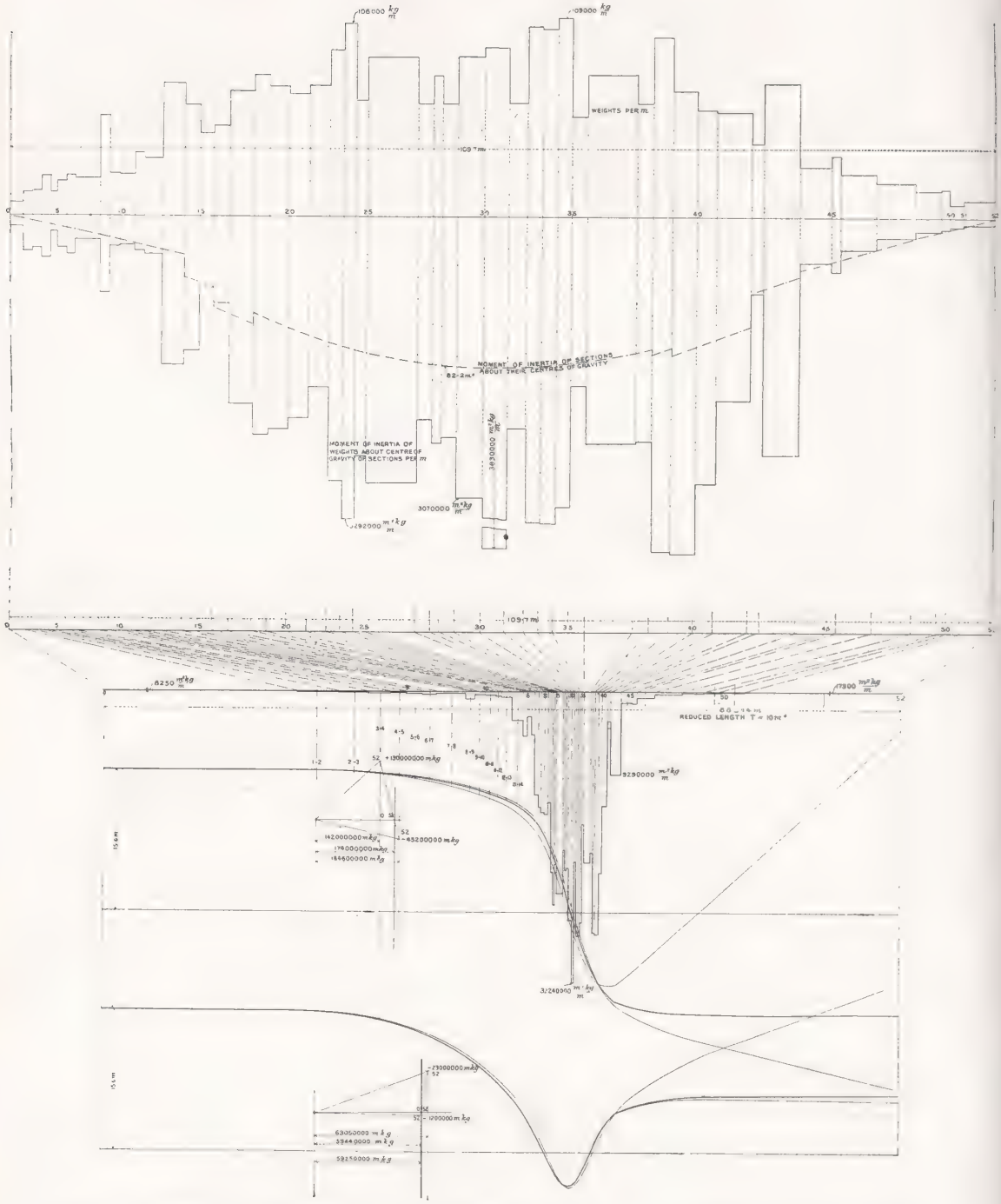

WEIGHTS PER m
MOMENT OF INERTIA OF SECTIONS ABOUT THEIR CENTRES OF GRAVITY
MOMENT OF INERTIA OF WEIGHTS ABOUT CENTRE OF GRAVITY OF SECTIONS PER m
REDUCED LENGTH
+130000000 m kg
-45200000 m kg
-23000000 m kg
-1200000 m kg

Fig. 2.—Method of obtaining Equivalent Shaft and Curve of Amplitudes of Vibrations.

with all necessary accuracy, by simple linear interpolation between them.

The frequency can be calculated directly from the magnitude of the polar distance $\frac{I\,G\,g}{\omega^2\,r^2}$ giving a frequency for the first order of free vibration of 650 and for the second order of 1,110 per min. The amplitudes thus determined for the equivalent shaft may be directly transferred to corresponding points of the ship, as shown in Fig. 3. The calculations prove that in the case of the *Vineta* torsional vibrations should not be set up by the engines, which make about 120 revolutions per min., and careful measurements on board have not revealed any such vibrations. It should be recognised, however, that the frequency of the torsional vibrations approaches the number of revolutions at which turbines run on similar vessels, and it may be expected that on turbine-driven ships the torsional vibrations will become more important than the vibrations due to bending.

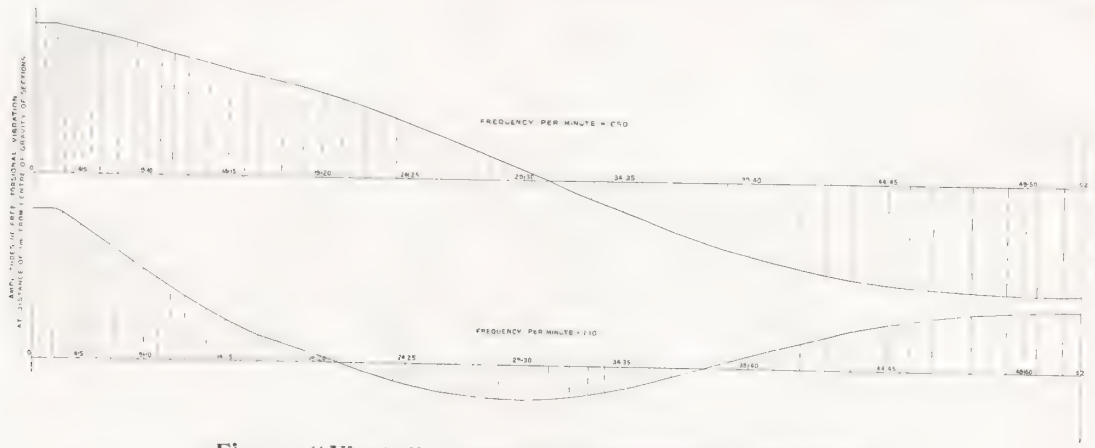


Fig. 3.—"Vineta": Curves of Amplitudes of Vibrations.

In conclusion, the author gives some general rules concerning the determination of vertical and torsional vibrations of ships. He states that for similar ships the critical speeds of revolutions are inversely proportional to the linear dimensions for both kinds of vibrations, or, in other words, the product of vibration frequency n and length of ship L is a constant for similar vessels.

In the case of fast steamers Schlick's measurements, calculated for the vertical vibrations, yield *e.g.*—

$n\,L$ = 13,800m. per minute = 45,300ft. per minute.

and for torsional vibrations—

$n\,L$ = 67,400m. per minute = 221,000ft. per minute.

In the case of the cruiser *Vineta*, the critical number of vertical vibrations has been measured as 136·5 per minute.

$n\,L$ = 13,900m. per minute = 45,600ft. per minute.

The critical number of torsional vibrations has been calculated to be 650.

$n\,L$ = 66,300m. per minute = 217,400ft. per minute.

Special types of ships must be separately considered. For instance, with torpedo boats and vertical vibrations—

$n\,L$ = 10,200m. per minute = 33,500ft. per minute,

and with torpedo boats and torsional vibrations—

$n\,L$ = 56,000m. per minute = 184,000ft. per minute.

Torsional vibrations of a detrimental character can be avoided by keeping the number of impulses (depending upon the revolutions of the engine and the number of propeller blades) about 15 per cent. away from the torsional frequency, and preferably 15 per cent. smaller.

When both vertical and torsional vibrations are to be suppressed, the number of impulses in the case of fast steamers should be kept—

$$\text{below}\ \frac{\cdot 85 \times 13800}{L}\ \text{or above}\ \frac{1{\cdot}15 \times 13800}{L}$$

$$\text{and below}\ \frac{\cdot 85 \times 67400}{L}\ \text{or above}\ \frac{1{\cdot}15 \times 67400}{L}$$

Hence, for a fast steamer of 150 metres between perpendiculars fitted with three-bladed propellers, the following numbers of revolutions of the engine should be avoided—

$$\frac{\cdot 85 \times 13800}{150}\ \text{to}\ \frac{1{\cdot}15 \times 13800}{150} = 78\ \text{to}\ 106.$$

$$\frac{\cdot 85 \times 67400}{150}\ \text{to}\ \frac{1{\cdot}15 \times 67400}{150} = 382\ \text{to}\ 516$$

and $\frac{382}{3}$ to $\frac{516}{3}$ = 127 to 173.

If the vessel is fitted with propellers with four blades, the two latter figures will become

$\frac{382}{4}$ to $\frac{516}{4}$ = 96 to 129.

A second way of suppressing, or at any rate of reducing, the torsional vibrations consists in erecting the engine which causes the vibration in the node of the curve of amplitudes of the natural vibrations.

A NEW VERTICAL VIBRATION RECORDER.

By Dr. S. Yokota.

(*Japanese Institution of Naval Architects; November*, 1911.)

IN this paper Dr. Yokota describes the principle and the working of his new vibration recorder, designed to provide an instrument handy in size and form, and at the same time accurate in its records. Its principal application is intended to be in connection with the vertical vibrations of steamers.

The principle of the recorder is as follows:— AB is a lever with a heavy weight at B. At the end A is an arm AE, fixed perpendicularly to AB. A spring is attached at E, pulling vertically downwards. Then, if the elongation of the spring when the lever AB is horizontal is equal to $\frac{l_2}{l_1}$ (see Fig. 1), the weight at B will remain approximately at rest for any small inclination of AB

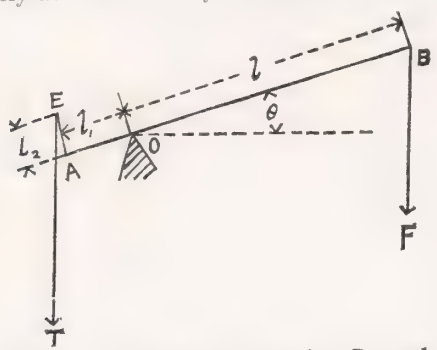


Fig. 1.—Principle of Vibration Recorder.

to the horizontal position; in other words the period of oscillation will be indefinitely long. The proof of this is as follows:—

Let W = the weight at B,

l = distance of W from C, the point of support,

T = tension of spring = ks, where s = extension of spring and k = a constant,

θ = (small) angle of displacement of the lever AB from the horizontal position.

Then the condition of equilibrium is

$$F + l\cos\theta = T(l_1\cos\theta + l_2\sin\theta)$$
$$= ks(l_1\cos\theta + l_2\sin\theta).$$

If this condition is fulfilled for varying values of θ, there is no impressed moment of force on the system.

If s_0 = elongation of spring when lever AB is horizontal

$$s = s_0 - l_2(1 - \cos\theta) - l_1\sin\theta.$$

Hence the condition of no impressed moment becomes

$$F + l\cos\theta = k(s_0 - l_2 + l_2\cos\theta - l_1\sin\theta)(l_1\cos\theta + l_2\sin\theta)$$

$$\text{or } F = k(s_0 - l_2 + l_2\cos\theta - l_1\sin\theta)\left(\frac{l_1}{l} + \frac{l_2}{l}\tan\theta\right)$$

Since θ is small, $\cos\theta = 1$

$$\sin\theta = \tan\theta = \theta$$

and $\theta^2 = 0$.

$$\text{Hence } F = k(s_0 - l_1\theta)\left(\frac{l_1}{l} + \frac{l_2}{l}\theta\right)$$

$$\text{and } \frac{F}{k} = s_0\frac{l_1}{l} + \left(s_0\frac{l_2}{l} - \frac{l_1^2}{l}\right)\theta.$$

To make this possible for varying values of θ, it is necessary that

$$s_0\frac{l_2}{l} - \frac{l_1^2}{l} = 0$$

$$\text{or } s_0 = \frac{l_1^2}{l_2}$$

VICE-ADMIRAL BARON AKAMATSU, I.J.N.,
President of the Japanese Institution of Naval Architects.

N. G. Akamatsu
Japanese Navy.

REAR-ADMIRAL CONSTRUCTOR M. FUKUDA,
Director of Naval Construction, Imperial Japanese Navy; and Member of Council,
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Dr. S. YOKOTA,
Professor of Naval Architecture, Tokyo Imperial University.

S. Yokota.

Dr. K. SUYEHIRO,
Professor of Naval Architecture, Tokyo Imperial University.

K. Suyehiro

末廣恭二

The Steering of Ships.

TURNING CIRCLES.

By Professor Wm. Hovgaard.

(*Institution of Naval Architects; March*, 1912.)

THE present paper contains an analysis of a number of turning trials of Danish and American warships of all classes, on the basis of which formulæ and coefficients are proposed by the author for an *a priori* determination of the diameter of the turning circle of a ship. The analysis is limited to the simple case of steady motion in a circle with the rudder at or near its maximum angle. All the results analysed, except one, were for vessels with twin screws and with rudders of ordinary form placed in their usual position relative to the screws; and in all cases, before the rudder was laid over, both propellers were going ahead at the same speed, and the valves of the engines were not touched during the trial.

Referring to Fig. 1, consider every point in

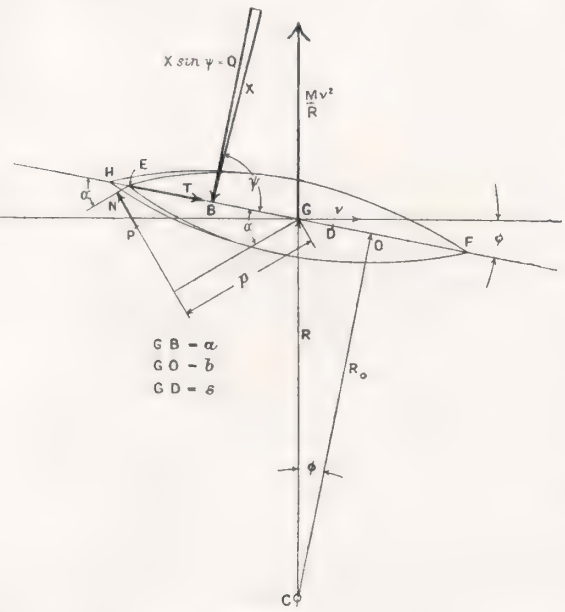


Fig. 1.

the ship to be moving with uniform velocity in a circle about a common centre, C, and let R equal the radius of the circle described by G, the centre of gravity of the ship. The translatory motion is maintained by the thrust of the propellers, T, acting along the axis of the ship. The angular motion is maintained by the normal pressure, P, on the rudder, assisted or hindered by a turning moment, L, due to unsymmetrical action of the propellers. The constant centripetal acceleration is resisted by the inertia of the mass, M, of the ship, which resistance is equivalent to a so-called centrifugal force $\frac{M v^2}{R}$ where v is the tangential velocity of G. All these forces are balanced by the water resistance X acting where shown on the diagram (Fig. 1). The component of X normal to the axis of the ship is denoted by Q. The axis of the ship is inclined at an angle, ϕ, known as the drift angle, to the tangent of the turning circle in which G is moving. The rudder angle is denoted by α and the normal pressure on the rudder (in tons) is determined by Joëssel's formula :—

$$P = \frac{\cdot 787\ A\ v^2 \sin \alpha}{2240\ [\cdot 195 - \cdot 305 \sin \alpha]} \qquad (1)$$

where A is the rudder area in square feet.

The value of v is found from :—

$$v = \frac{\pi R}{t} \text{ ft. sec.} \qquad (2)$$

where t is the time for turning through the last half circle.

Let the point of application of the normal rudder pressure be N, then its leverage about the centre of gravity of the ship is :—

$$p = GE \cos \alpha + EN \qquad (3)$$

The equations of motion may then be written down as follow :—

$$Q = \frac{M v^2}{R} \cos \phi + P \cos \alpha \qquad (4)$$

$$X \cos \psi = T - P \sin \alpha - \frac{M v^2}{R} \sin \phi \qquad (5)$$

and taking moments about G

$$Qa = Pp + L \qquad (6')$$

where L is a turning moment due to inequality in the thrust of the starboard and port propellers.

As L may be disregarded in all ordinary cases, the simple formula is obtained—

$$Qa = Pp \qquad (6)$$

The trials upon which the analysis depends were those for which the drift angle was observed. From the trial results the value of the rudder pressure, P, and the centrifugal force, $\frac{M v^2}{R}$, were calculated, after which the lateral component of the resistance, Q, and its leverage, a, about the centre of gravity were found by means of formulæ (4) and (6). In some cases the propeller thrust, T, could be estimated with fair accuracy, and the longitudinal component of the resistance, $X \cos \psi$, could then be found from formula (5). Since $Q = X \sin \psi$ had already been calculated, X and ψ could then be determined. The resultant resistance, X, was found to be the dominant force and to act very nearly at right angles to the centre-line of the ship. It is much greater than either the rudder pressure or the propeller thrust, and is expended chiefly in overcoming the centrifugal force.

To solve the problem in a complete manner, it is necessary to express Q and a in terms of known quantities. For thus expressing Q the author proposes a coefficient K_1 which is obtained by the formula :—

$$K_1 = \frac{QR}{v^2 S (b - s)} \qquad (7)$$

where

S = Area of centre-line plane exclusive of rudder.

$s = GD$ = Distance of centre of gravity, D, of centre-line plane forward (+) or aft (−) of centre of gravity of ship.

$b = GO = R \sin \phi$ = Distance of pivoting point forward of centre of gravity of ship.

By means of this formula K_1 was calculated for all the trials for which ϕ was known and the results plotted in ratios of $\frac{\text{Breadth at } WL}{\text{Mean draught}}$ as shown in Fig. 2.

For expressing a, a coefficient K_2 is proposed, of which the value is determined by the formula :—

$$K_2 = \epsilon \frac{\delta a (b - s)}{[\rho^2 - s (b - s)]} \qquad (8)$$

where $\epsilon = \frac{S}{LD}$ is the coefficient of fineness of the centre-line plane, L being the length and D the draught of the ship ; ρ is the radius of gyration of the centre-line plane about a vertical through its centre of gravity D ; and δ is the prismatic coefficient of fineness of the vessel. The coefficient K_2 is intended to take account of the general form of hull and is plotted in Fig. 3 from the trial results on the ratios $\frac{\text{area of midship section}}{\text{area of centre-line plane}}$ as abscissæ.

The determination of the curves for K_1 and K_2 completes the analysis, and by their use the radius of the turning circle of a ship may be predicted with such approximation as this method and the experimental material on which the analysis is based will permit. The steps to be taken are the following :—

The principal dimensions and other necessary data are collected or calculated. Thereafter K_1 and K_2 are found from the curves, and $(b - s)$ is found from :—

$$b - s = \frac{\epsilon \rho^2 K_2 \cos a + \frac{\delta p M \cos \phi}{K_1 S}}{\delta p - \epsilon s K_2 \cos a} \qquad (9)$$

which is derived from the foregoing formulæ by elimination of v and R. The only unknown quantity in this expression is ϕ, which has to be assumed provisionally. By means of the value of b determined from equation (9), a more accurate value of ϕ is obtained, and, if necessary, $(b - s)$ is re-calculated. The radius of the turning circle is then determined from :—

$$R = \frac{K_1 S (b - s) - M \cos \phi}{A \cos a (\cdot 0018 \sin a + \cdot 00115)} \qquad (10)$$

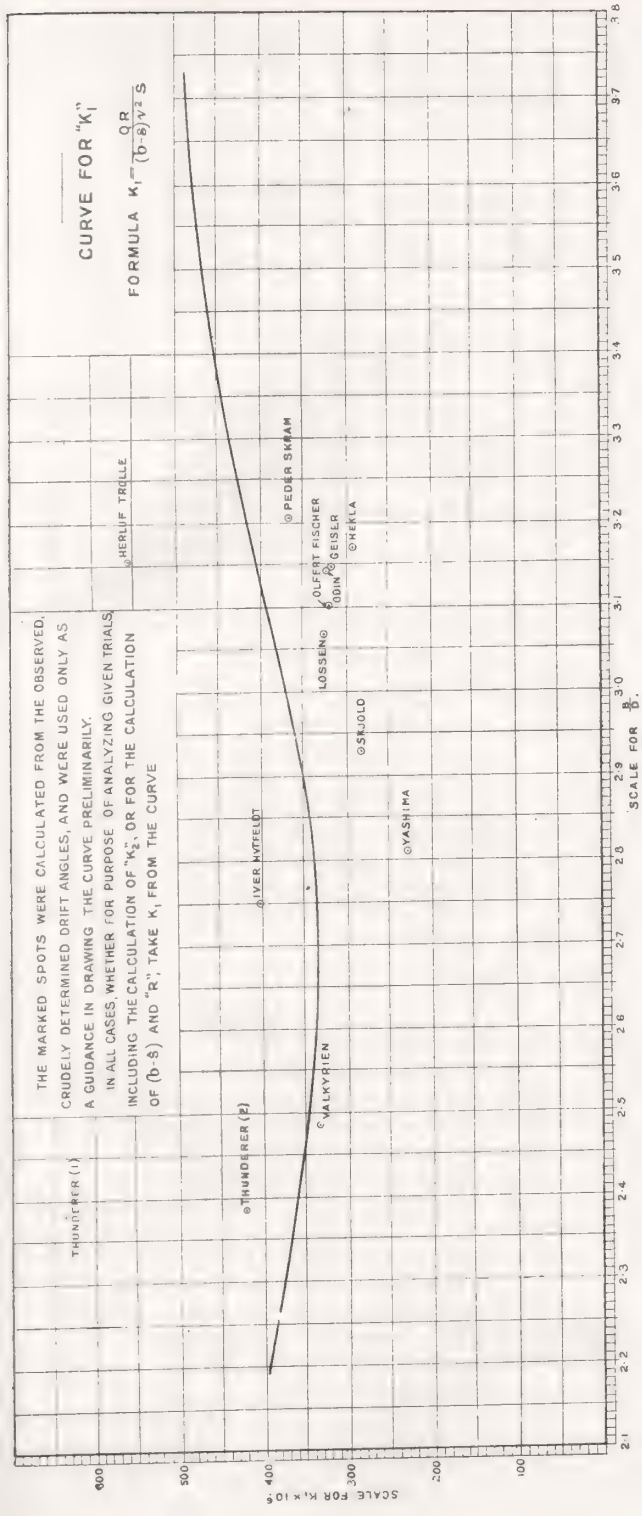
which is derived from formulæ (1) and (4) by substituting the value of Q from (7).

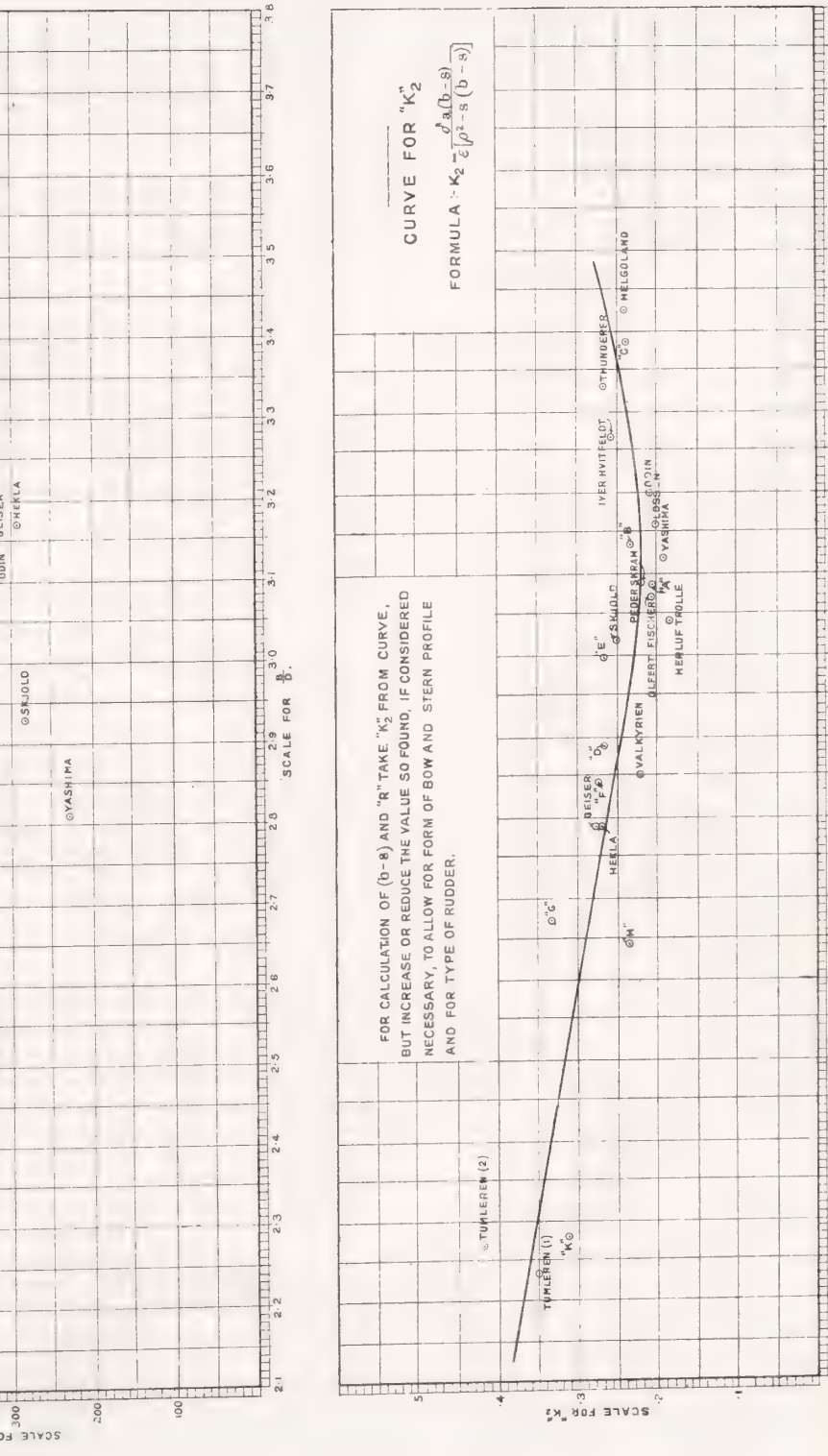
Finally, the author considers some special turning trials made with Danish warships, and makes the following statements :—

I.—In cases where the speed of the ship was increased during the trial from zero to a certain maximum, the radius of the turning circle was found to be only slightly greater than at uniform speed.

II.—In cases where the ship was turned by one engine going ahead at a uniform speed with the other engine stopped, the rudder being amidships, it is shown by the formulæ that the turning effort, to which the ship must have been subject in order to make it turn in the observed circle, was far in excess of that which could be directly caused by the propeller thrust, and that the greater part of the turning moment can only be accounted for by the suction on the quarter in front of the propeller, which suction is not, as when both propellers are going ahead, balanced by suction on the other side of the ship.

III.—For trials similar to II. but with the rudder hard over instead of amidships, it was


CURVE FOR "K1"
FORMULA K1 = Q R / (b-s) v² S
THE MARKED SPOTS WERE CALCULATED FROM THE OBSERVED, CRUDELY DETERMINED DRIFT ANGLES, AND WERE USED ONLY AS A GUIDANCE IN DRAWING THE CURVE PRELIMINARILY. IN ALL CASES, WHETHER FOR PURPOSE OF ANALYZING GIVEN TRIALS INCLUDING THE CALCULATION OF "K2", OR FOR THE CALCULATION OF (b-s) AND "R", TAKE K1 FROM THE CURVE
THUNDERER (1)
HERLUF TROLLE
PEDER SKRAM
OLFERT FISCHER
ODIN
GEISER
HERLA
LOSSEN
SKJOLD
YASHIMA
IVER HVITFELDT
VALKYRIEN
THUNDERER (2)
SCALE FOR K1 x 10⁵
SCALE FOR b/D.


CURVE FOR "K2"
FORMULA K2 = δ² a (b-s) / ε [ρ² - s (b-s)]
FOR CALCULATION OF (b-s) AND "R" TAKE "K2" FROM CURVE, BUT INCREASE OR REDUCE THE VALUE SO FOUND, IF CONSIDERED NECESSARY, TO ALLOW FOR FORM OF BOW AND STERN PROFILE AND FOR TYPE OF RUDDER.
TUMLEREN (1)
TUMLEREN (2)
HERLA
GEISER
VALKYRIEN
SKJOLD
PEDER SKRAM
OLFERT FISCHER
LOSSEN
YASHIMA
HERLUF TROLLE
IVER HVITFELDT
THUNDERER
HELGOLAND
SCALE FOR K2
SCALE FOR R

Figs. 2 (at top) **and 3** (at bottom).—**Values of K_1 and K_2.**

found by analysis that the action of the rudder, when counteracting the turning effect of the propeller whereby it was subject to the direct impact of the propeller race, was at least three times as great as when the rudder was turned away from the propeller race, and nearly one and a half times as great as when both propellers were used.

INFLUENCE OF THE DIRECTION OF ROTATION OF THE SCREWS IN A TWIN-SCREW SHIP UPON THE TURNING OF THE SHIP FROM A STATE OF REST.

By Herr Direktor M. Walter.

(*Schiffbautechnische Gesellschaft; November*, 1911.)

THE author's investigation into this subject resulted from the adverse reports which were received with respect to the manœuvring of two relatively small twin-screw passenger vessels, the *Vorwaerts* and the *Gazelle*, belonging to the North German Lloyd, which in their service have to navigate very narrow waterways. The particulars of the two vessels, which were provided with inward turning propellers, are as follow :--

Vorwaerts.

Length B.P.	205ft. 4in.
Breadth at waterline	31ft. 2in.
Depth moulded	12ft. 11in.
I.H.P.	1,000
Revolutions per minute	135
Speed	12 knots.
Two four-bladed propellers of 8ft. 7in. dia. and 11ft. 10in. pitch.	

Gazelle.

Length B.P.	105ft. 0in.
Breadth at deck	17ft. 0in.
Depth moulded	7ft. 6in.
I.H.P.	270
Revolutions per minute	230
Speed	12 knots.
Two three bladed propellers of 4ft. 3in. dia. and 8ft. 3in. pitch.	

Both these vessels followed earlier sister ships which were provided with outward-turning screws. The fact that in manœuvring qualities the two later ships were found to be inferior led to the inference that the adverse results were due to the change in the direction of rotation of the propellers. An explanation of the matter was then sought by theoretical considerations. The turning movement of a vessel from a state of rest is effected by running one engine ahead and the other astern. For example, when it is required to turn the vessel to starboard the starboard engine is set astern and the port engine ahead. If the vessel has inward-turning screws, then the reversal of the starboard engine makes the starboard screw turn outwards and both port and starboard propellers will be revolving in the right-hand direction, as shown in Fig. 1. The starboard propeller will then draw water away from one side of the ship, while the port propeller will heap water against the other side, the reactions set up tending to reduce the desired

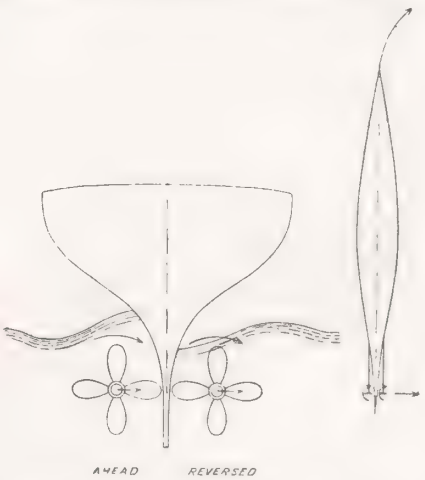


Fig. 1.—Inward-turning Screws.

movement of the stern to port. On the other hand, if the vessel has outward-turning screws, when the starboard engine is reversed both screws will revolve in a left-hand direction, and the reaction of the water upon the ship's side, as shown in Fig. 2, will tend to increase the desired movement. As the steamer *Vorwaerts* had a large aperture in way of the propellers, however, it seemed unlikely that the poor steering qualities of the steamer were entirely due to the above influence, and a further cause was looked for.

It is a well known fact that in the case of a single-screw vessel with a right-hand propeller starting from rest, the bow moves to port and the stern to starboard, as shown in Fig. 3, and *vice versâ* with a left-hand propeller. The explanation which has been put forward for this action is that the lower blades are working in solid water and

hence experience a greater resistance than the upper blades which are working in broken water near the surface. Consequently, there is a force equal to the difference between the upper and lower resistance, tending, in the case of the right-hand propeller, to move the stern to starboard. With twin propellers this effect will be doubled, and will tend to further reduce the turning moment in the case shown in Fig. 1 (inward-turning screws) or tend to increase the turning moment in the case shown in Fig. 2 (outward-turning screws). From the above considerations the author concludes that outward-turning screws are the best for manœuvring from a position of rest. When the ship is under way and turning, the influence of the propellers upon the turning moment is only small in comparison with that exerted by the rudder, and the direction of rotation of the propellers is of no practical importance.

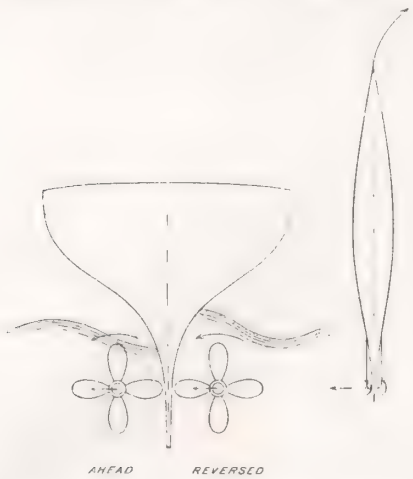


Fig. 2.—Outward-turning Screws.

Guided by the foregoing conclusions, it was decided to interchange the port and starboard crankshafts and screws of the *Vorwaerts*. This was done in December, 1906, and has been attended with the most satisfactory results. A further improvement was effected in 1910 by plating in the aperture. The time required to completely turn the ship was reduced from about $4\frac{1}{2}$ before to $3\frac{1}{2}$ minutes after the alterations were made. The success of the alterations to the *Vorwaerts* led to a similar change from inward to outward-turning screws being made in the case of the *Gazelle*, with equally satisfactory results.

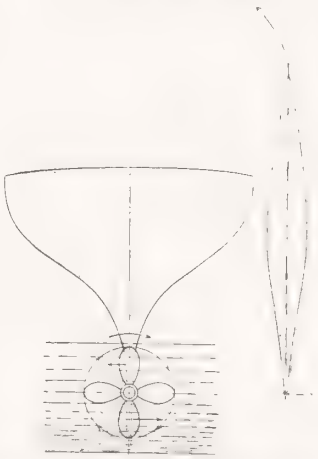


Fig. 3.—Single Right-hand Screw.

WINDLASS AND STEERING-GEAR ENGINES.

By Professor F. P. Purvis.

(*Institution of Engineers and Shipbuilders in Scotland, November*, 1911.)

THIS paper contains proposed rules for determining the size of the cylinders of windlass engines and of steering-gear engines. The rule for windlass engines is :—

$D = 5d$ for, say, 100 lb. pressure per square inch,

where D = diameter of cylinders,

d = diameter of material of chain cable.

To obtain this relation Professor Purvis makes the following assumptions, *viz.* :—

Holding power of anchor varies as weight of anchor varies as d^2,

Ratio of gearing the same in all cases,

Diameter of cable lifter varies as stroke, so that speed of lift varies as stroke $\times$ revolutions.

Then, since the work to be done per revolution varies as holding power $\times$ diameter of lifter, and work of engine varies as $D^2 \times$ stroke, it follows that

$$D^2 \propto d^2$$
$$\therefore\ D \propto d$$

The multiplier 5 is, of course, obtained from experience.

The proposed rule for steering-gear engines is:

$D^{2\frac{1}{2}} = \frac{1}{60} \times L \times$ draught,

where D = diameter of cylinders in inches,

L = length B.P. of ship in feet,

Draught = mean load draught in feet.

The justification of this rule is somewhat complex, and is given in an appendix to the paper. In obtaining the rule the author assumes that resistance to turning is independent of form of ship and varies only with dimensions. Along with this assumption there are several others of a similar nature, one being that the distribution of weights is similar in different ships. It is obvious that in many ships the conditions of these assumptions are not complied with, but it is considered that the error involved by making them will be comparatively small. The rule also involves a relation between time of turning and length of ship such that, at a definite speed, the time of turning through a definite angle should vary approximately as the length of ship. Another assumption is that piston speed varies as $D^{\frac{1}{2}}$. On this assumption it follows that the power of the engine varies as $D^{2\frac{1}{2}}$, the quantity on the left-hand side of the equation.

From this rule Table I. is derived. In a few cases comparison with actual present-day practice is quoted to justify the value of the constant adopted, *viz.*, $\frac{1}{60}$.

TABLE I.

$L \times$ Draught.	Diameter of Cylinders by Rule. Inches.	Ship's Name.	$L \times$ Draught.	Actual Cylinders. Inches.	Maker of Gear.
1,500	4				
2,600	5				
3,900	6				
5,600	7				
7,700	8				
10,100	9	*Hakata Maru*	11,350	$\frac{8,8}{10}$	Caldwell
12,900	10	*Hirana Maru*	12,600	$\frac{11,11}{10}$	Caldwell
16,100	11				
19,800	12				
23,800	13				
28,300	14	*Mauretania*	27,500	$\frac{14,14}{14}$	Brown
33,300	15	*Olympic*	29,300	$\frac{17,17,17}{18}$	Harland & Wolff
38,700	16			in duplicate	Harland & Wolff
44,600	17				
51,000	18				

ELECTRICAL STEERING.

By Mr. B. P. HAIGH, B.Sc.

(*British Association; September*, 1911.)

ELECTRICAL steering offers considerable advantages as regards economy for steamers as well as for vessels propelled by internal-combustion engines, although it is in connection with the latter, owing to the absence of steam power, that it is receiving most attention. The main problem of steering gear design is to obtain a reliable system of control, capable of dealing

with the power necessary to put the helm hard over in the shortest possible time in case of emergency, and also possessing sufficient sensitiveness to enable an accurate course to be kept by moving the rudder promptly in small angles. In the author's opinion both these requirements can be better fulfilled with an electrical gear than with the usual steam gear.

Sensitiveness is shown by absence of "time-lag" between the movement of the hand-wheel and the corresponding movement of the rudder. This quality also requires an absence of undue "idle travel" of the hand-wheel. A certain small amount of idle-travel is nevertheless desirable to prevent the gear being used unnecessarily. In the highest class work the steering gear should respond to a motion of the hand-wheel corresponding to one degree of helm, or less than one-eighth turn of the wheel. In steam gears the idle travel is kept down by reducing the already small steam-lap on the control-valve, which, in order to obtain a fast-running gear with little time-lag, is generally a piston valve of ample size. Such a valve is liable to leakage when slightly worn, thus causing a waste of power in the steam gear, to which there is no corresponding loss in the case of the electrical gear.

The steering motor may be started and stopped for every motion of the rudder, but it is preferably kept running continuously, mechanical control being introduced either in the form of hydraulic transmission or magnetic clutches as developed by the author. In the latter type of gear two magnetic clutches are employed, these being fitted at opposite ends of the motor; and as no gearing is kept continuously in motion, the wear and tear, as well as the current required, are reduced to a minimum. The clutches prevent the shock of the sea being transmitted to the electrical system; and as they have considerable flywheel effect, the current taken by the motor does not fluctuate widely under normal conditions, and the steering gear may therefore be supplied from the ship's lighting generator. To economize power it is advantageous to arrange the gear so that greater leverage is obtained when the rudder is hard over than when amidships, and by doubling the leverage in this manner a saving of 30 per cent. may be made in the motor power. Drawings are given in the paper of a steering gear suitable for an 11½-inch rudder-post, and of a small gear of the same type, built by Messrs. Brown Brothers & Co., Ltd., of Rosebank, Edinburgh, suitable for a 7-inch post. When tested against an artificial hydraulic load, the latter gear developed a torque of 50 foot-tons at the rudder-post and showed an efficiency of over 50 per cent. at half load. It was found capable of moving the tiller through 70° in 23 seconds and responded to motions of the hand-wheel equivalent to 1° of helm.

Floating Docks.

THE RAISING OF THE DRY DOCK "DEWEY."

By Naval Constructor L. S. Adams, U.S.N.

(*American Society of Naval Architects and Marine Engineers; November*, 1911.)

THE raising of the dry dock *Dewey*, which sank at Olongapo, Subic Bay, Philippine Islands, in May, 1910, was a salvage operation somewhat out of the ordinary. The *Dewey* is a steel floating self-docking dry dock, with a lifting capacity of about 20,000 tons and dimensions as shown in Fig. 1. The light weight of the dock is about 11,000 tons. The central pontoon is divided into 14, and each end pontoon is divided into 5 main watertight compartments. These main compartments are still further subdivided by watertight bulkheads, so that there are 32 watertight compartments in the main pontoon and 18 in each end pontoon, making 68 in all. Combined flooding and drainage pipes with quick-acting gate valves lead from each of these 68 compartments to a main drain and flooding pipe running along in the bottom of the port side of the dock. The valves in each main compartment are all operated by one lever in the valve house. The pumping machinery consists of three 24-in. steam-driven horizontal centrifugal pumps and three Babcock & Wilcox boilers. The flooding of the ballast compartments is accomplished from sea valves connected to the main drain.

On the afternoon of the 23rd May, 1910, the dock was lowered to a depth of 26ft. of water over the keel blocks, to be in readiness for docking two torpedo-boat destroyers next day. During the night the dock gradually sank, the cause at that time being unknown. On the 24th May the top deck of the side wall on the port side was 6ft.

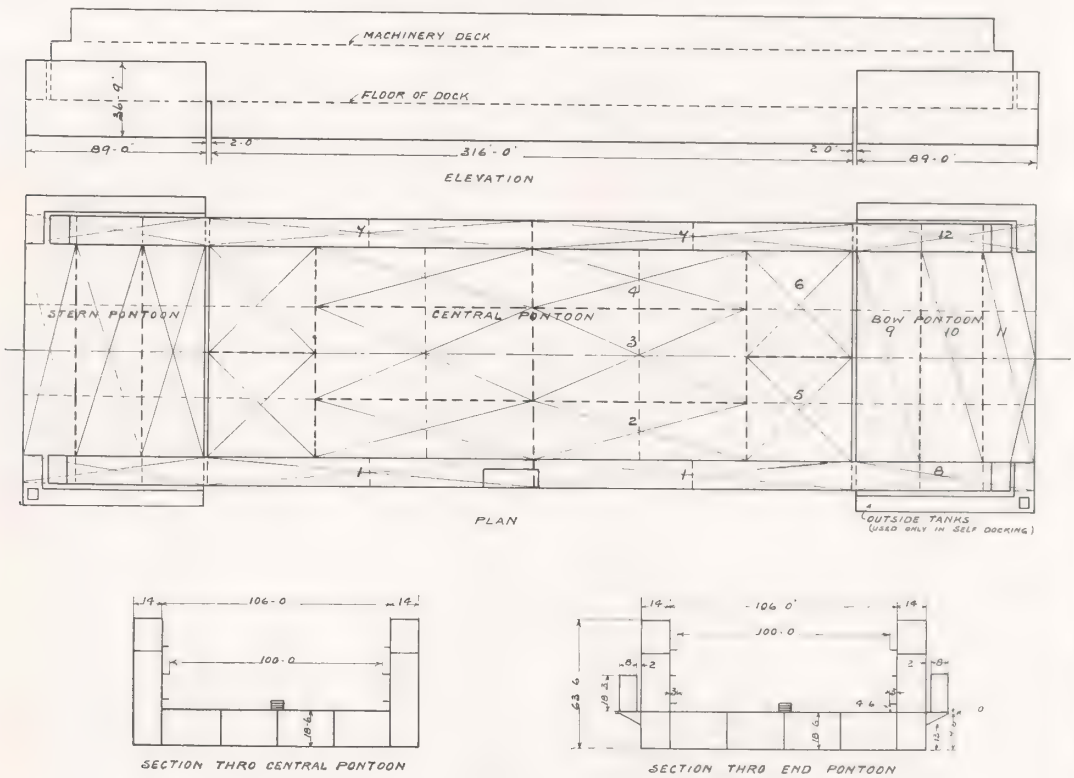


Fig. 1.—Floating Dry Dock "Dewey."

Fig. 2.—Condition at 9-20 a.m., 26th May, 1910.

Fig. 3.—Condition at 5-15 p.m., 14th June, 1910.

Fig. 4.—Condition at 10-45 a.m., 15th June, 1910.

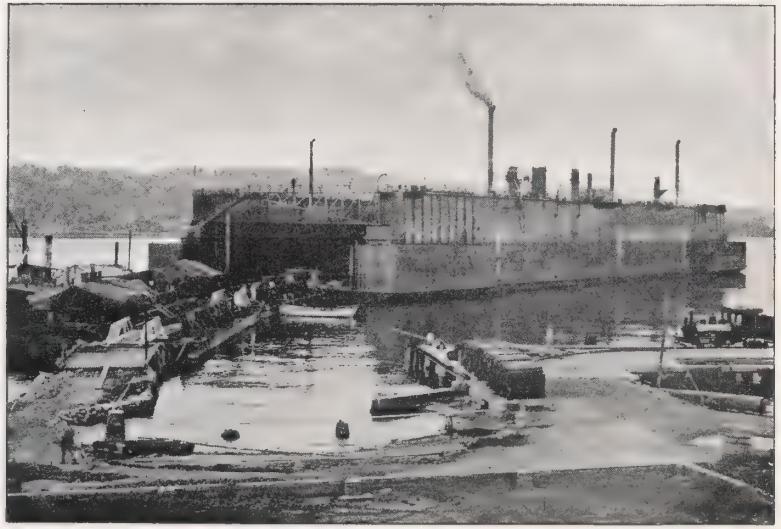


Fig. 5.—Condition at 10-10 a.m., 29th June, 1910.

under water at the forward end and about 16ft. under water at the after end at mean tide level. The starboard side at first remained afloat, but was gradually filling with water. Work was immediately begun to save this side by closing all openings in the top deck, and five small pumps were mounted in the machinery spaces to pump water from No. 1 tank up through the manholes. The state of affairs on the 26th May is shown in Fig. 2. The starboard side had then gone down, so that its outer end was about 3ft. under water. By this time, however, the pumps mentioned above were in operation and the dock was prevented sinking further.

As an examination of the dock by divers showed that there was no serious damage to the structure and no large opening to the sea, the alternative plans determined upon for attempting to raise the dock were as follow :—

(1) By introducing compressed air into bottom tanks Nos. 4, 6, 9, 10, 11 and 12 on the port side.

(2) By making watertight the compartment in which the forward centrifugal pump for operating the dock was located ; then clearing this compartment of water and operating the dock's own pump by steam furnished from the tug *Wompatuck*, or by running this pump under water by means of compressed air.

(3) By making the whole top and bottom decks and the side walls of the machinery space on the sunken side watertight ; then clearing this whole space of water, and finally using the dock's own pump to raise her. This was the surest method, but the longest and most costly.

In pursuance of the first method, air connections were made to the tanks on the low side, but it was found that the air went at once to the tanks on the high side owing to the tank bulkheads being non-watertight at the top, a fact unknown before this attempt was made. This method, therefore, proved to be impracticable and even dangerous, and was abandoned.

With regard to the second method, the air pressure available was found insufficient to work the dock's own pump, but by the 14th June the latter was put into operation by steam from the *Wompatuck*, with the first result shown in Fig. 3. The starboard side came up first, the pump taking water from the tanks in that side before it would take any water whatever from the tanks in the low side, no matter which valves were opened or closed. This action was wholly unlooked for, and the operation was stopped temporarily until the No. 1 tanks on the high side had been completely filled with water and the vents plugged. The dock's pump was then again put into operation until the dock reached the position shown in Fig. 4.

It was considered dangerous to pursue this second method any further, owing to the large quantity of water in the forward end pontoon and the consequent risk of straining the dock's structure. The third method of raising the dock, for which preparations had been in progress, was then begun. Five cofferdams were fitted for access to the pumps and valves. After the water in the machinery spaces had been removed by means of pumps mounted in the cofferdams, the dock's own pump was eventually started on the 29th June, and the dock finally raised without any further difficulty ; see Fig. 5.

As soon as the dock was raised a careful examination was made to determine the cause of sinking. It was found that the ventilation pipes from the tanks were badly corroded, in many cases being reduced to a mere shell, which had given way and thus left large areas open to the sea. When the dock is up these air pipes are always above the surface of the water ; but when the dock had been lowered to take the torpedo-boat destroyers, the air pipes were mostly below water, and the water entering them had gradually filled the ballast compartments of the dock, thus causing it to sink.

THE MARINE TERMINAL OF THE GRAND TRUNK PACIFIC RAILWAY, PRINCE RUPERT, BRITISH COLUMBIA.

By Mr. F. H. Kirby and Mr. Wm. T. Donnelly.

(*American Society of Naval Architects and Marine Engineers ; November*, 1911.)

THE authors describe the ship-repairing and shipbuilding plant which is to be installed at Prince Rupert, B.C., for the Grand Trunk Pacific Railway Company. The installation will include a floating dock of 20,000 tons lifting capacity, so designed as to be capable of operating in sections as a number of smaller docks, and an adequate shore plant, comprising electric power generating plant with air compressors, a machine shop, boiler and blacksmiths' shops, and a covered construction shed under which the pontoons of the floating dry dock will

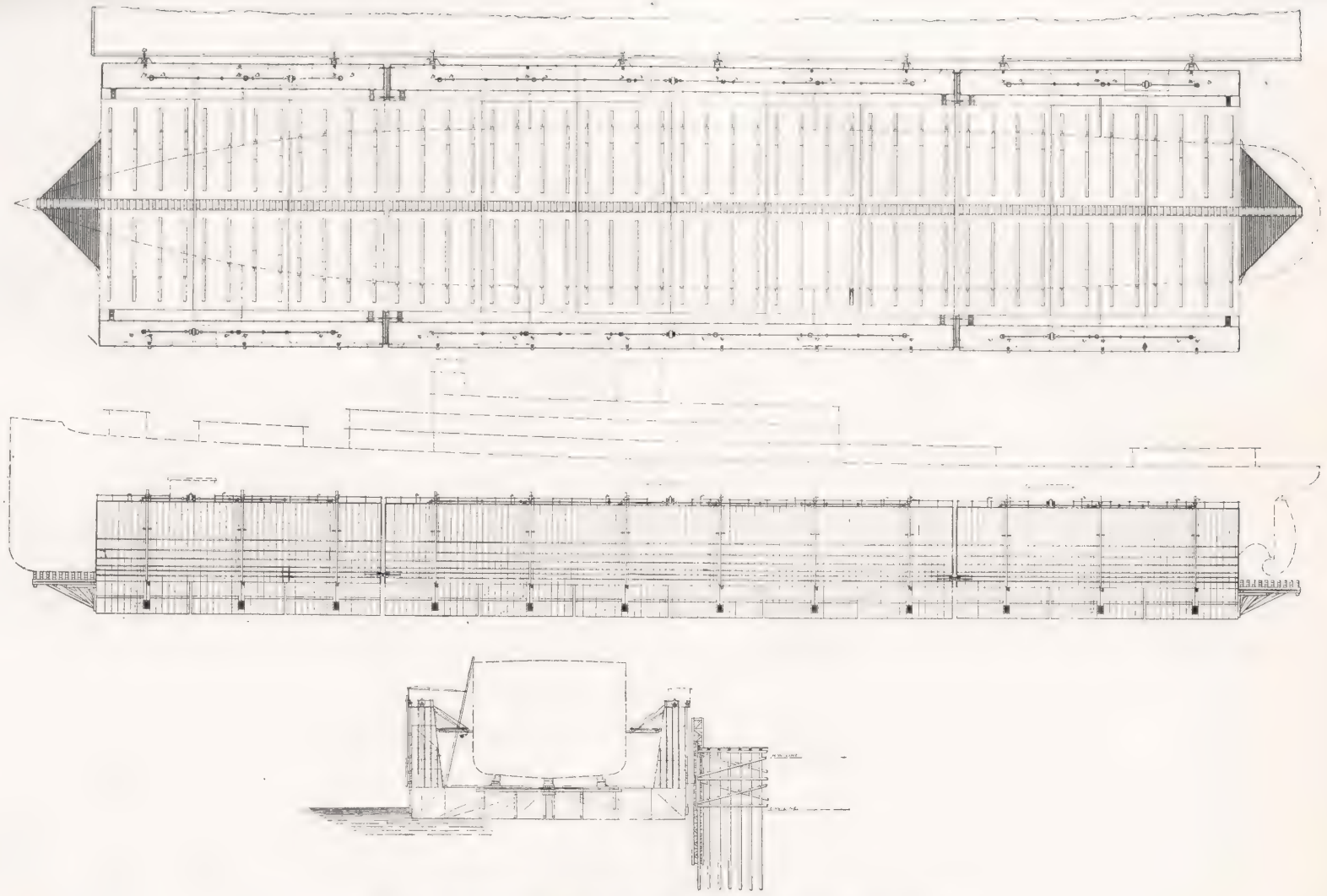


Fig. 1.—20,000-ton Floating Dry Dock for Prince Rupert, B.C.

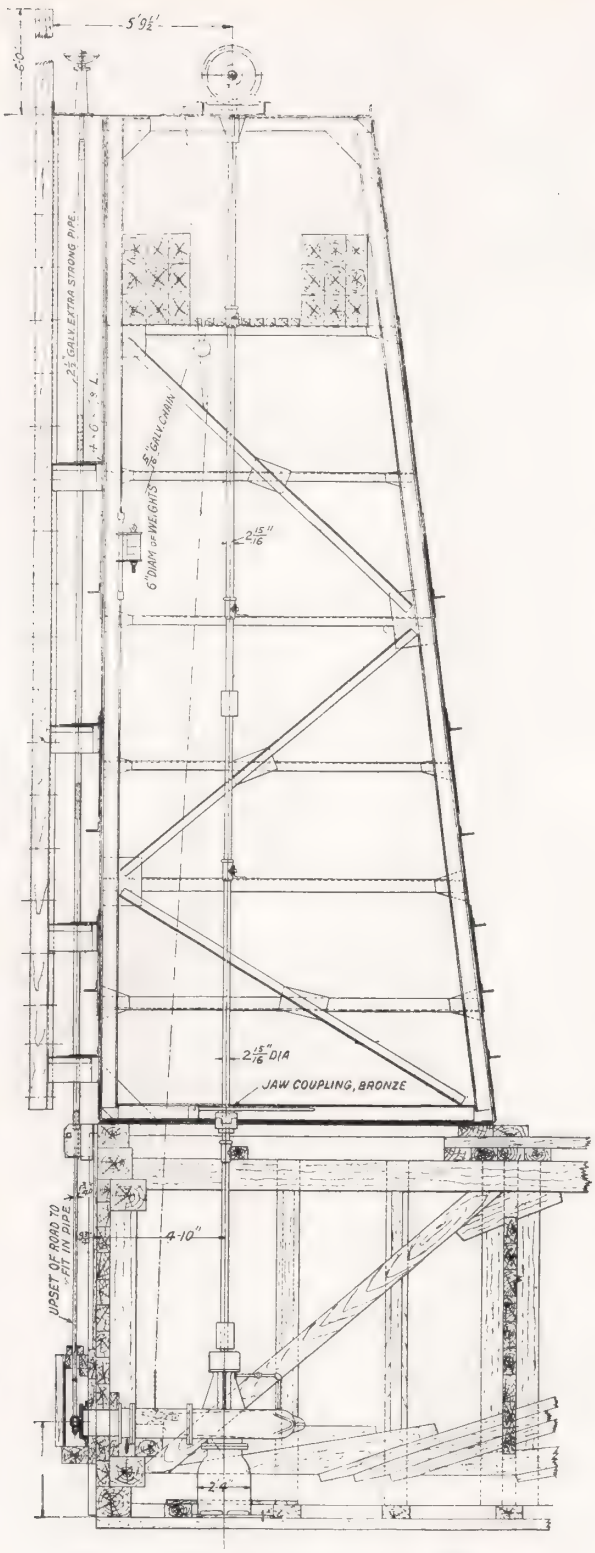


Fig. 2.—Section through Side Wall.

be built. The general plan provides for the practical completion and equipment of the shore plant on the most modern lines before the floating dock is commenced, so that the latter can be almost entirely built upon the site, due account having been taken of the fact that the city of Prince Rupert is 600 miles from the nearest base of supply. The dock, of which a general arrangement is given in Fig. 1, is to have an overall length at the keel blocks of 604ft. 4in., a clear width of 100ft., and a width overall of 130ft. The lifting power is provided by twelve timber pontoons 130ft. wide, 44ft. long, and 15ft. deep. The pontoons will be attached to steel side walls 38ft. high, 15ft. wide at the bottom, and 10ft. wide at the top, the walls being divided so that the whole structure may be used under ordinary conditions as three separate docks, one of six pontoons with an overall length of 269 ft., and two of three pontoons with an overall length of 164ft. each. The structure as a whole is secured to the shore by the engagement of clamps on the dock, with a vertical truss secured to the pile platform or pier in such a way that the dock is free to rise and fall with the tide and when being raised or lowered for docking purposes.

The construction of the timber pontoons is shown in Fig. 3. Each pontoon has 15 trusses spaced on 3-ft. centres. The centre-line water-tight bulkhead is 12in. thick, and there are three fore and aft 6-in. partial bulkheads on each side of the centre-line. All diagonal braces are heavily reinforced with anchor stocks. The arch brace is made up of planking through-bolted with screw bolts, and is intended to take the reverse stresses when the dock is floating light. Beams 6in. by 12in. are worked across the upper and lower truss members to carry the 5-in. deck and bottom planking.

To protect the exterior from toredo and other marine worms, it is first thoroughly graved with tar poisoned with arsenic; then sheathed with two layers of hair felt, each thoroughly saturated with tar and arsenic; and then with creosoted lumber, also treated with arsenic and thoroughly secured with galvanized nails.

The steel side walls (Fig. 2) are formed by channel and angle frames and bracing 3ft. apart, covered by plating varying in thickness from $\frac{1}{2}$in. to $\frac{5}{16}$in. Where the side wall meets the deck of the pontoon there is a steel shoe secured to each pontoon frame and to each wall frame. These

shoes are connected together by a steel link, 15in. long, and pins, the upper pin being tapered. The driving of this pin wedges the pontoon and side wall together. At the point of contact the bottom of the wall is reinforced by a 12-in. × ½-in. plate and made watertight by canvas packing saturated with red lead.

The pumping machinery consists of twenty-four 12-in. centrifugal pumps driven by vertical shafts and electric motors, and is capable of pumping the entire lifting power of the dock in less than two hours. Indicators will be provided to show the level of water in the different compartments, and to automatically control the inflow of water should the dock get out of the level.

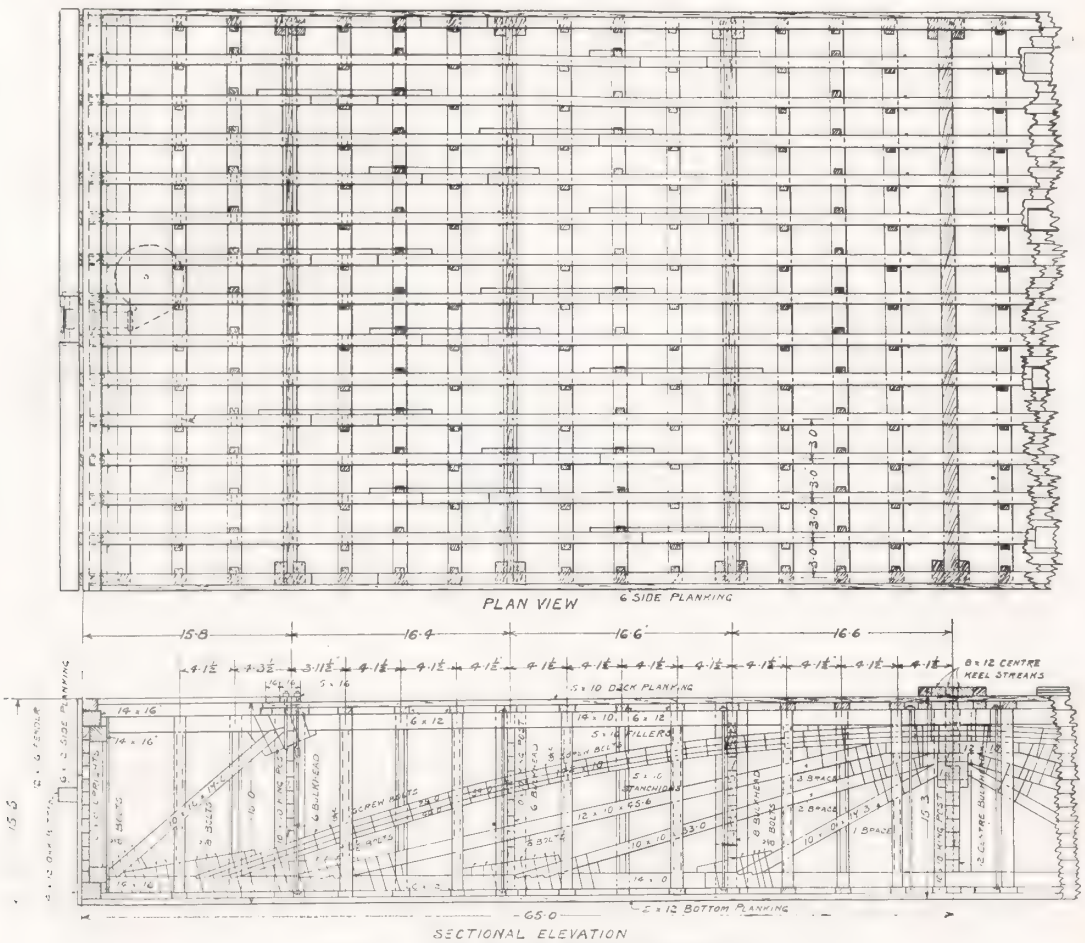


Fig. 3.—One of the Timber Pontoons.

Marine Boilers.

RESULTS OF EXPERIMENTS WITH A WATER-TUBE BOILER, WITH SPECIAL REFERENCE TO SUPERHEATING.

By Mr. Harold E. Yarrow.

(*Institution of Naval Architects; March*, 1912.)

THE author discusses some important experiments which Messrs. Yarrow & Company recently carried out with a water-tube boiler of the Yarrow type fitted with a special form of superheater. Throughout the experiments oil fuel only was used, and careful records were taken of the oil consumed, the water evaporated, steam pressure, temperature of the superheated steam, and the temperature of the gases at various points during their passage past the boiler tubes. Fig. 1 shows a cross section of the boiler experimented with. On the side where the superheater is placed there are fewer rows of generator tubes than on the other side, it being thought desirable that the total heating surface and the resistance to the gases on both sides of the boilers should be approximately the same. The heating surface on the superheater side of the boiler is 3,453 square feet, of which 1,265 square feet is superheating surface, and on the other side 3,247 square feet, giving a total heating surface of 6,700 square feet.

The superheater consists of a number of U tubes expanded into two longitudinal collectors. The placing of the superheater on one side of the boiler only is the leading feature of the arrangement, and a damper is provided in the uptake on the same side, so that the superheater may be shut off should the engines be suddenly eased or stopped or when steam is being raised, in order to prevent damage to the tubes or excessive superheat due to insufficient circulation of steam.

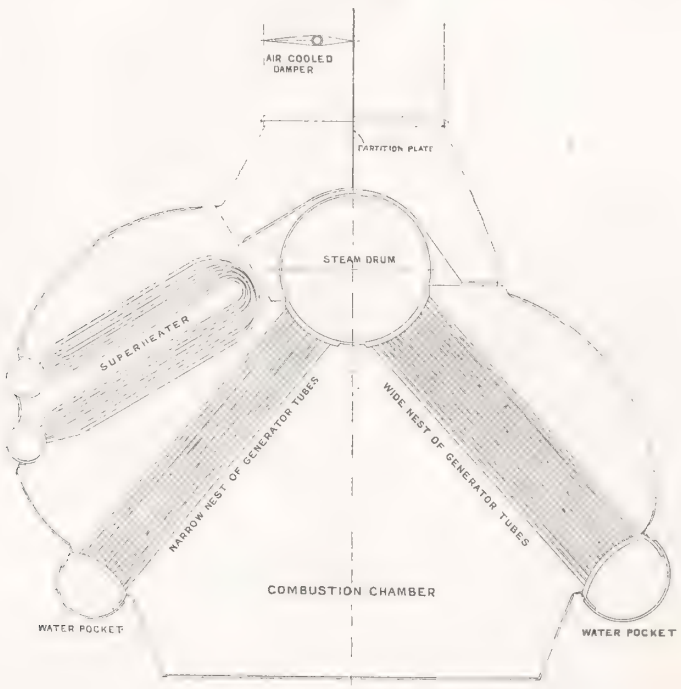


Fig. 1.—Cross Section of Yarrow Boiler with Superheater, experimented with.

Table I.

Trials with Damper Open.

Heating Surface { Large Nest of Generator Tubes = 3,247 square feet; Small Nest of Generator Tubes = 2,188 square feet; Superheater = 1,265 square feet } 6,700 square feet total.

On these trials the heating surface is taken as the total heating surface of 6,700 square feet.

Steam Pressure, lb. per square inch.	Superheat in degrees Fahrenheit.	Air Pressure, inches of Water.	Lb. of Water Evaporated per hour.	Lb. of Oil Fuel Burnt per hour.	From and at 212 degrees Fahrenheit.		Lb. of Oil Fuel Burnt per square foot of heating surface per hour.	Temperature of Feed Water, degrees Fahrenheit.	Temperature between Small Nest of Generator Tubes and Superheater, degrees Fahrenheit.	Temperature of Uptake, degrees Fahrenheit.	
					Lb. of Water Evaporated per lb. of Oil per hour.	Lb. of Water Evaporated per square foot of heating surface per hour.				Above Superheater.	Above Large Nest of Generator Tubes.
242	93·5	5·0	94,659	8,286	14·6	18·0	1·237	58·0	1,121	828	887
243	93·0	3·16	76,021	6,454	15·0	14·4	·9635	63·5	926	698	727
243·7	82·5	2·44	68,387	5,695	15·2	12·9	·850	63·5	903	685	688
242·8	61·1	1·7	46,041	3,630	15·9	8·6	·542	64·0	647	536	551
241·8	31·0	·998	20,059	1,540	16·1	3·7	·230	62·2	481	432	448
242·2	20·75	·625	8,478	649	16·1	1·55	·096	63·5	465	409	416

Table II.

Trials with Damper Shut.

Heating Surface { Large Nest of Generator Tubes = 3,247 square feet; Small Nest of Generator Tubes = 2,188 square feet; Superheater = 1,265 square feet } 6,700 square feet total.

On these trials the heating surface of Boiler is taken as heating surface of Large Nest of Generator Tubes = 3,247 square feet.

Steam Pressure, lb. per square inch.	Air Pressure, inches of Water.	Lb. of Water Evaporated per hour.	Lb. of Oil Fuel Burnt per hour.	From and at 212 degrees Fahrenheit.		Lb. of Oil Fuel Burnt per square foot of heating surface per hour.	Temperature of Feed Water, degrees Fahrenheit.	Temperature of Uptakes, degrees Fahrenheit, above Large Nest of Generator Tubes.
				Lb. of Water Evaporated per lb. of Oil per hour.	Lb. of Water Evaporated per square foot of heating surface per hour.			
242·0	4·85	68,648	6,287	13·25	25·66	1·936	61·0	913
242·25	3·97	57,693	5,065	13·84	21·6	1·56	60·0	843
242·4	2·491	44,050	3,504	15·3	16·5	1·09	60·3	673
242·5	1·46	31,481	2,473	15·4	11·75	·76	63·5	603

A further advantage of this arrangement is that it greatly diminishes the output of the boiler at the time when a reduced supply of steam is wanted.

These experiments also suggested another important improvement in the Yarrow boiler. It was discovered that leakages which had been experienced at the seams of the water pockets

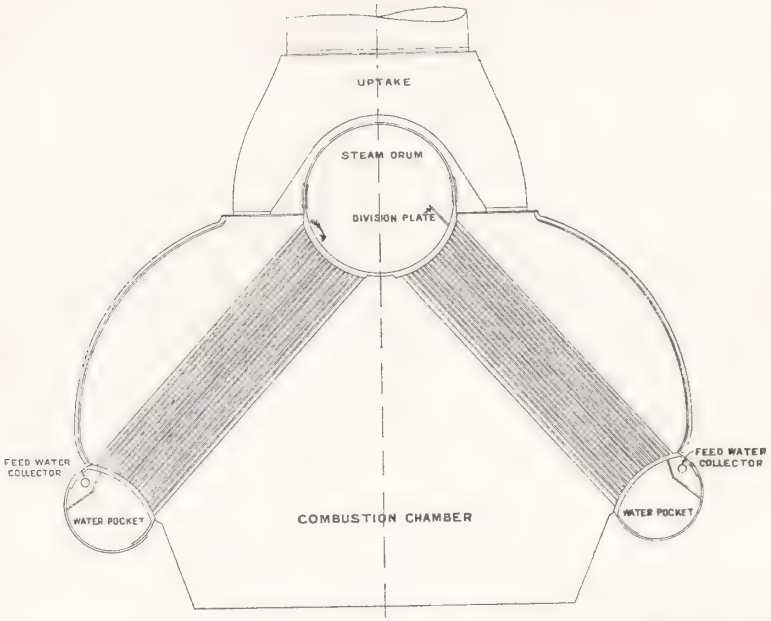


Fig. 2.—Cross Section of Yarrow Boiler showing Division Plate to prevent Short-circuiting of Feed.

The results of two series of trials, with and without the superheater in use respectively, are given in Tables I. and II. In the case of the trials recorded in Table II, advantage was taken of the opportunity to make tests burning oil fuel at a rate of consumption considerably greater than has hitherto been the custom. These and other experiments indicate that, in a properly designed boiler of the type dealt with, it is possible to burn, without injury to the boiler, 2 lb. of oil per square foot of heating surface per hour.

Boilers fitted with superheaters have been installed by Messrs. Yarrow on one of the destroyers built by them for the British Admiralty —H.M.S. *Archer*—and on the official trials of this vessel the expected gain due to superheating was fully realized. On the full-speed trial the degree of superheat at the turbines was 94° F. The S.H.P. developed was slightly over 18,500, which compares with about 17,000, the S.H.P. the builders would have expected had the boilers been of the usual type. The mean speed obtained on the six runs on the measured mile at Skelmorlie was 30·9 knots, and the mean speed for eight hours 30·3 knots, the contract speed being 28 knots.

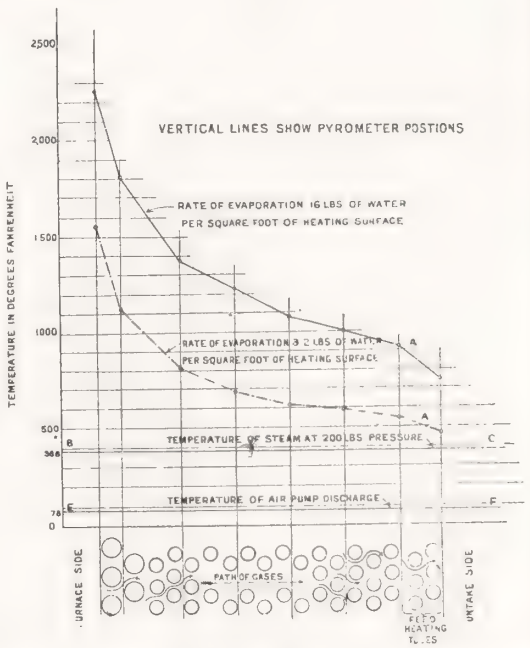


Fig. 3.—Curves showing Drop of Temperature of Flue Gases.

when working boilers of the Yarrow type at high rates of evaporation were due to sharp changes of temperature, caused by short-circuiting of the feed water, as indicated by the arrow on the left-hand side of Fig. 2. It was found that by simply placing longitudinal division plates in the upper chamber, as shown on the right-hand side of Fig. 2, so as to avoid the short-circuiting of the feed, all difficulties disappeared.

The temperature of the gases at various points in the experimental boiler, as found by pyrometers, are shown in Fig. 3. From this diagram will be seen the very great drop in temperature which takes place during the passage of the gases through the first rows of tubes, showing the large proportion of heat that is taken out of the gases by these tubes. It will also be observed that there is a sudden drop in the temperature at A and A^1, where the gases pass through the last rows of tubes, in which the cold feed water is ascending. This fact suggests the desirability of extending the feed-heating system, as shown in

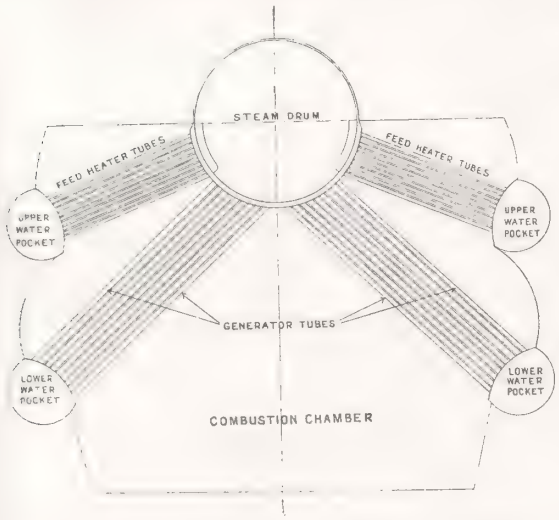


Fig. 4.—Yarrow Boiler with separate Feed-heating Tubes and Water Collectors.

Fig. 4, by having separate water collectors and feed-heating tubes, apart from the main water collectors and main generator tubes.

The author then describes three arrangements for superheating and feed-heating which may be adopted with a view to still further improving the results in connection with such a boiler as the one dealt with. The design shown in Fig. 5 will probably be the most efficient for a given quantity of heating surface.

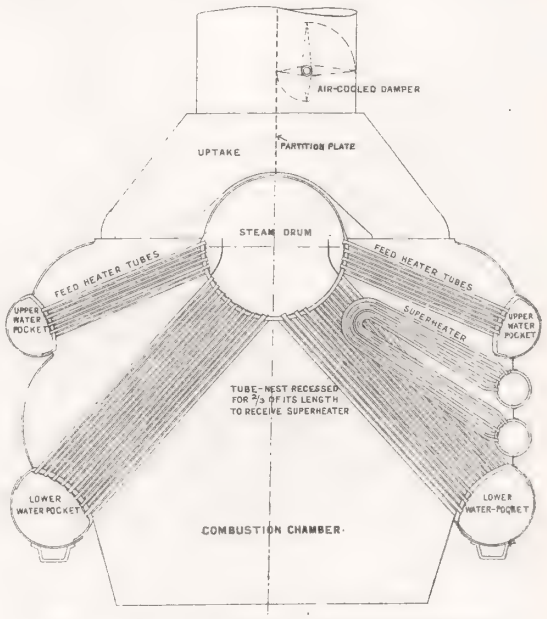


Fig. 5.—Yarrow Boiler with Feed-heaters and Superheater.

In conclusion, Mr. Yarrow submits that, from the results of the accumulated experience of others and from the experiments described, there will be a certain gain by the use of superheated steam of from 8 to 10 per cent. in fuel economy when using 100° F. of superheat, and from 11 to 13 per cent. gain when using 150° F. of superheat in combination with a pressure of 200 lb. per square inch. A further gain in fuel economy can also be obtained by an efficient system of heating the feed from the gases after they have passed the generator tubes. With regard to the relative weights of boilers, with and without superheaters and with equal heating surfaces, there would be no appreciable difference. If the complete machinery installation is taken into account, there would probably be a small saving in weight in the case of the installation with superheated steam.

NEW TYPE OF ASH EXPELLERS ON THE U.S.S. "CYCLOPS."

By Mr. J. F. Metten.

(*American Society of Naval Engineers; November*, 1911.)

THIS paper contains a description of a new type of under-water ash ejector installed on the 20,000-ton naval collier *Cyclops*. Fig. 1 shows a cross section through the expeller, the essential feature of which consists of a tapered tube connecting a hopper located above the boiler-room floor to the bottom of the ship. The lower part of the tube is formed with an annular chamber contracted at the lower end to form an ejector nozzle, a pump connection being provided above the inner bottom. At two points in the tube revolving gates are fitted, operated by a vertical shaft connecting to a crank at the hopper end. These gates are of circular form, and each is provided with a single opening of the same diameter as the tube. The openings are spaced so that the lower gate is open when the crank is in one extreme position against the stop, and the upper gate is open when the crank is in the other extreme position. The water is supplied through the annular nozzle under pressure, so that the kinetic energy of the issuing jet is considerably more than that necessary to overcome the static pressure at the mouth of the opening due to the waterline head.

The pressure required on the expellers of the *Cyclops*, operating at the load draught of 28ft., is about 60 lb. per square inch, and is supplied by a small DeLaval turbine-driven centrifugal pump. Assuming the lower operating gate closed and the upper open, ashes are shovelled into the hopper until the tube fills nearly to the upper gate. The handle is then swung quickly against the opposite stop, thus closing the upper and opening the lower gate, which allows the contents to drop by gravity into the vortex of the annular jet and to be discharged into the sea. The ejectors have been in use on the *Cyclops* over a year without requiring any parts renewed and are reported as showing no appreciable signs of wear.

Extensive experiments have recently been carried out with a view to ascertaining the best shape of nozzle, and a form of annular jet has been determined, with which it has been found possible to operate an expeller of the *Cyclops* type having a 7-in. diameter tube under a head of 30ft., with 27 water horse-power and a pressure of 50 lb. per square inch, a result which shows considerably less power than required to operate an ejector of the ordinary type.

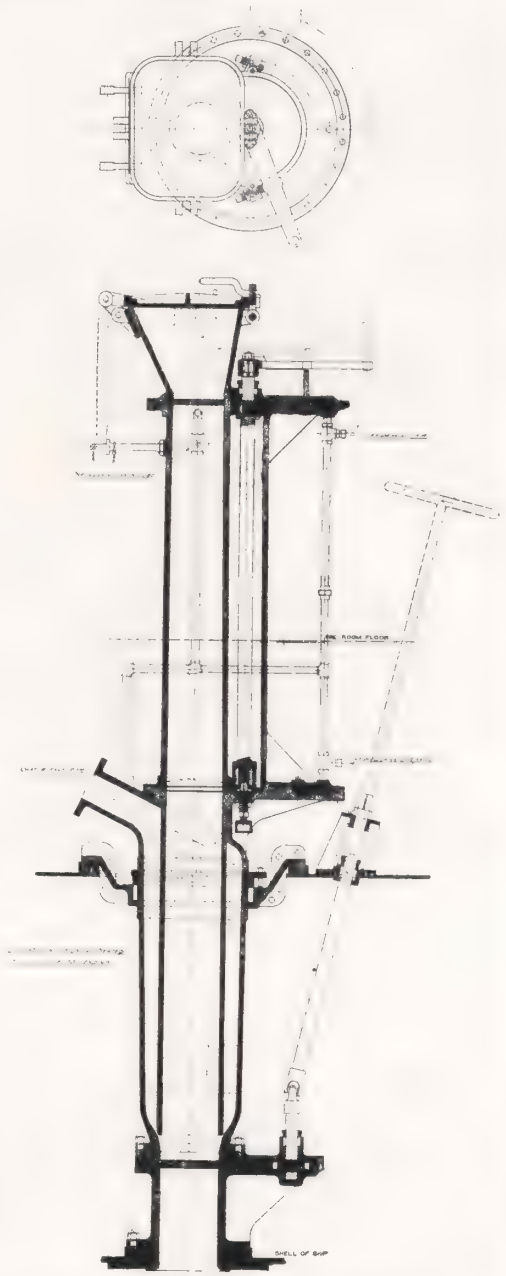


Fig. 1.—Ash Expeller, U.S.S. "Cyclops."

THE SOLIGNAC-GRILLE BOILER AND ITS APPLICATION IN FRENCH CHANNEL STEAMERS.

By Monsieur G. Hart.

(*Institution of Naval Architects ; March*, 1912.)

THE principle adopted for circulating the water and passing the steam through the tubes in the Solignac-Grille boiler is based upon the introduction at the orifice of the tube, on the feed-water side, of a supplementary resistance to prevent the reversals in the direction of flow of either fluid, which are frequently experienced in marine boilers having tubes slightly inclined to the horizontal, and to lessen the tendency of the steam to escape in jerks and in the form of large bubbles.

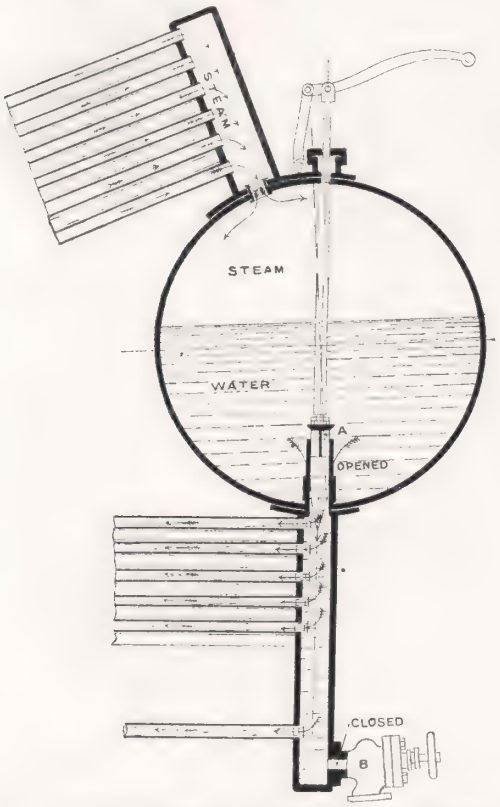


Fig. 1.—Normal Circulation in Solignac-Grille Boiler.

This supplementary resistance consists of a diaphragm fitted at the lower end of the tube, which prevents any escape of steam at that end and at the same time limits the flow of water into the tube to the quantity corresponding, after regulating the inlet orifice, to the quantity of water which can be vapourized. The normal circulation in the boiler is shown in Fig. 1. As the escape of steam takes place above the water level in the boiler, the production of dry steam is assured and priming is done away with, provided that the opening of the diaphragm is well under control.

Another advantage of the Solignac-Grille boiler is the facility for cleaning the inside of the tubes with a jet of steam, under pressure, proceeding in the reverse direction to the normal circulation. This operation is accomplished while

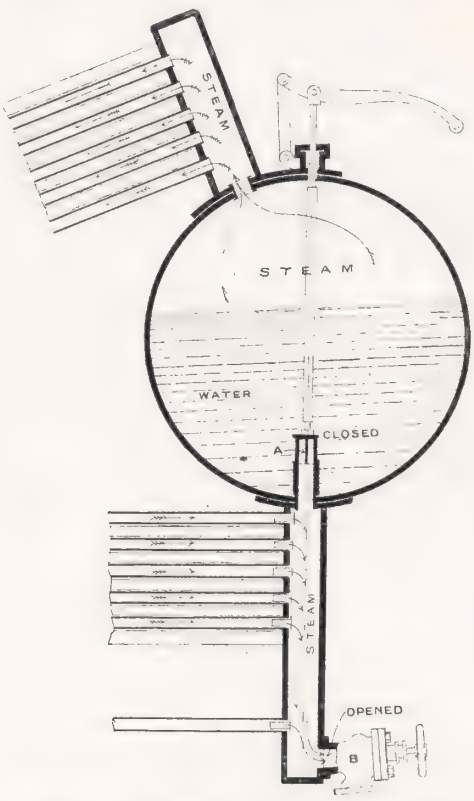


Fig. 2.—Method of Cleaning Tubes in Solignac-Grille Boiler.

the boiler is under steam and under fire in the rapid and simple way indicated by Fig. 2, the quickness of the operation ensuring that the tubes are not deprived of water for sufficient time to become damaged by the effect of heat radiation.

Trials on an experimental boiler of the Solignac-Grille type were carried out in Paris in 1901 and

at Loughborough (England) the following year. Subsequently more extensive trials on full-sized boilers were made at Indret and Calais Maritime in 1903 and 1905, and as a result in 1908 the French Admiralty decided to fit two Solignac-Grille boilers in the torpedo boat *Kabyle*. This installation gave very satisfactory results and enabled a considerable saving in weight and space to be effected. In 1908–9 four boilers were fitted, jointly with eight Lagraffel and d'Allest boilers, on board the channel steamer *Pas de Calais* belonging to the French Northern Railway Co., which had previously been equipped with twelve boilers of the latter type. The principal dimensions and particulars of the two types of boilers as fitted in this vessel are given in Table I, from which it will be seen that the Solignac-Grille type has the advantage both in weight and space occupied.

One of the Solignac boilers is illustrated in Fig. 3.

The four Solignac-Grille boilers working with a rate of combustion of 200 kgr. per square metre of grate correspond to six Lagraffel and d'Allest boilers working at 132 kgr. The coal consumption is 2,795 kgr. for the former and 3,120 kgr. for the latter, which represents a saving of about 12 per cent., despite the difference in the rate of combustion, which, with the highest rate of combustion, causes a reduction in the evaporative power per kgr. of coal. The evaporation should, therefore, be much better in the Solignac-Grille boilers than in the Lagraffel and d'Allest. That this is so was proved on the actual service of the *Pas de Calais*, the steam consumption, although more steam was available, being less by 8 per cent. than when the vessel was fitted with d'Allest boilers only.

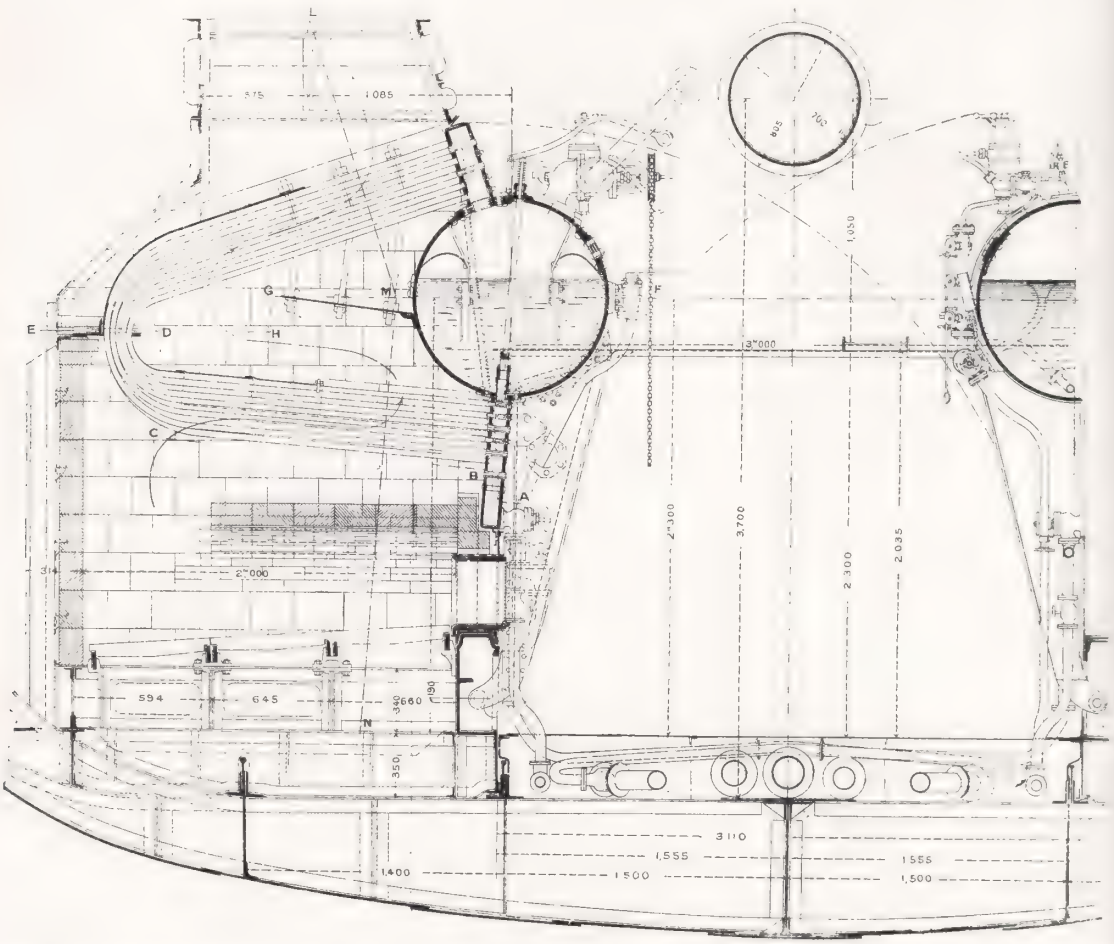


Fig. 3.—One of the Solignac-Grille Boilers in Channel Steamers "Le Nord" and "Pas de Calais."

Owing to the success of the experiment with the *Pas de Calais*, the *Le Nord*, belonging to the same owners, had her entire installation of Lagraffel boilers replaced by twelve Solignac-Grille boilers. This vessel has now re-entered her service and attained a speed notably greater than that of the *Pas de Calais*. The combustion rate is 166 kgr. per square metre of grate, and the pressure (16 kgr. per square centimetre) is easily maintained, the level of the water being almost constant. The time for raising steam with the Solignac boilers is one-half that required with Lagraffel boilers; and when the ship starts with a low pressure and with fires in a bad state, in consequence of the late arrival of trains, the normal pressure is obtained in four or five minutes, while ten or twelve minutes were necessary with the Lagraffel boilers.

TABLE I.

Particulars of the Boilers.	Lagraffel and d'Allest.	Solignac-Grille.
Working pressure, kgr.	15	15
„ length, m.	2·00	1·932
Grate width, m.	2·00	1·806
„ area, sq. m.	4	3·249
Tubes, number	259	258
„ internal diameter, m.m.	62 & 64	25
„ external diameter, m.m.	70	30
Mean length of the tubes, m.	2·200	4·421
Heating surface in contact with the hot gases, sq. m. — direct	5·90	2·563
Heating surface in contact with the hot gases, sq. m. — tubular	125·35	107·20
Heating surface in contact with the hot gases, sq. m. — total	131·25	109·76
Ratio of heating surface to grate area	32·8	31·44
Volume of water, cub. m.	3·200	1·300
Volume of steam, cub. m.	2·380	0·913
Height of the chimney above the grate, m.	17	17
Section of chimney corresponding to one boiler, sq. m.	0·625	0·625
Required floor space, sq. m.	6·80	5·98
Weight in working order (water included), tons	17·5	13

THE POSSIBILITIES OF FLUE GAS ECONOMIZERS ON BOARD SHIP.

By Mr. R. ROYDS, M.Sc., and Mr. J. W. CAMPBELL, M.Sc.

(*Institution of Engineers and Shipbuilders in Scotland; January*, 1912.)

THIS paper describes some experiments which have been made by the authors at the Glasgow and West of Scotland Technical College. The arrangement of the apparatus used is shown in Fig. 1. Air under a small pressure was passed through the gas-heater and then allowed to flow through the inner tube, while a flow of water was maintained in the annular space between the inner and outer tubes. The principal results are shown in Figs. 2 to 5.

Fig. 2 refers to the tests made with different rates of air-flow at different air-inlet temperatures. In Fig. 3 the heat transmission per square foot, expressed per degree Fahrenheit temperature difference between the air and water, has been plotted in relation to the weight of air-flow, and it will be seen that the inlet temperature of the air had only a minor influence on the rate of heat transmission per degree Fahrenheit difference. Fig. 4 refers to the tests made under nearly constant conditions as regards weight of air-flow, the weight of water-flow being varied with different constant temperatures of the inlet air. It will be seen that the influence of the velocity of the water on the rate of heat transmission is practically negligible when compared with the influence of the air velocity.

of flow at inlet + Loss of energy at inlet − Pressure energy recovered from kinetic energy at outlet. If care is taken with the form of the outlet connections, the authors' experiments showed that nearly ·4 of the kinetic energy at exit could be recovered.

The authors then proceed to show how the heat transmission results obtained from the experiments could be applied to the design of an economizer. By accepting that Reynolds' law of heat transmission is a general law, and from consideration of the fact that in any flue gas economizer on board ship the gas flow would need to be high, whereas, as shown by these experiments, the velocity of the water need not have more than a moderate value, the authors finally arrive at the following expression :—

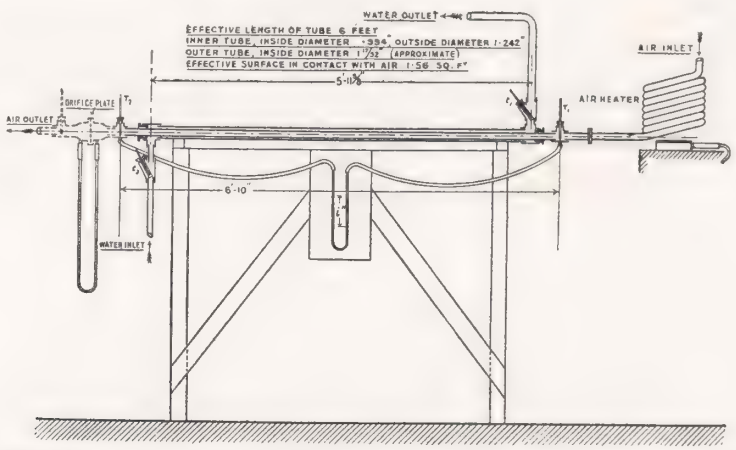


Fig. 1.—Arrangement of Apparatus used for the Experiments.

The values obtained for the tube resistance to the flow of air were found to obey the following law :—

$$\frac{i \times p}{\tau} = q\, w^2$$

as represented in Fig. 5, where

i = pressure drop through the tube in inches of water.
p = arithmetical mean pressure in tube, lb. per square inch.
τ = arithmetical mean temperature in tube, degrees Fahrenheit absolute.
w = weight of air-flow, lb. per minute.
q = a constant = ·02128.

This expression only relates to the resistance of the tube, and takes no account of the end resistances. Had the tube been connected to large inlet and outlet spaces, as would obtain under working conditions, the total resistance would be expressed by :—Resistance of tube + Pressure required to produce the kinetic energy

$$\frac{l}{m_1} = \frac{k_1\, K}{K-1} \times \frac{c_1 + C_2\, R\, \beta}{c_1\, C_2\, R\, \beta} \times \log_e \frac{T_1 - t_1}{T_2 - t_2}$$ in which

l = length of economizer tubes, in feet.
m_1 = $\dfrac{\text{sectional area of tube}}{\text{perimeter of tube}}$ in feet.
T_1 = temperature of gases at inlet, in degrees Fahrenheit.
T_2 = temperature of gases at outlet, in degrees Fahrenheit.
t_1 = temperature of feed-water at outlet, in degrees Fahrenheit.
t_2 = temperature of feed-water at inlet, in degrees Fahrenheit.
k_1 = specific heat of gases = ·24.
K = $\dfrac{T_1 - T_2}{t_1 - t_2}$
β = $\dfrac{\text{outside diameter of tube}}{\text{inside diameter of tube.}}$
R = $\dfrac{a_1}{w_1}$

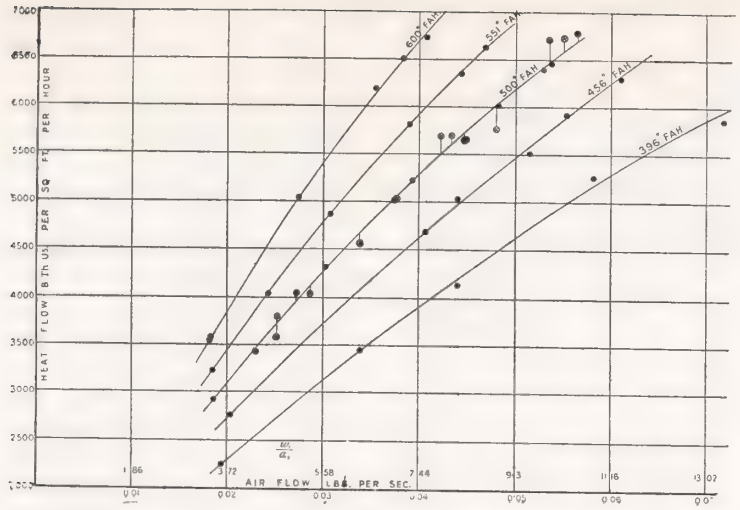


Fig. 2.—Tests with Different Rates of Air-Flow.

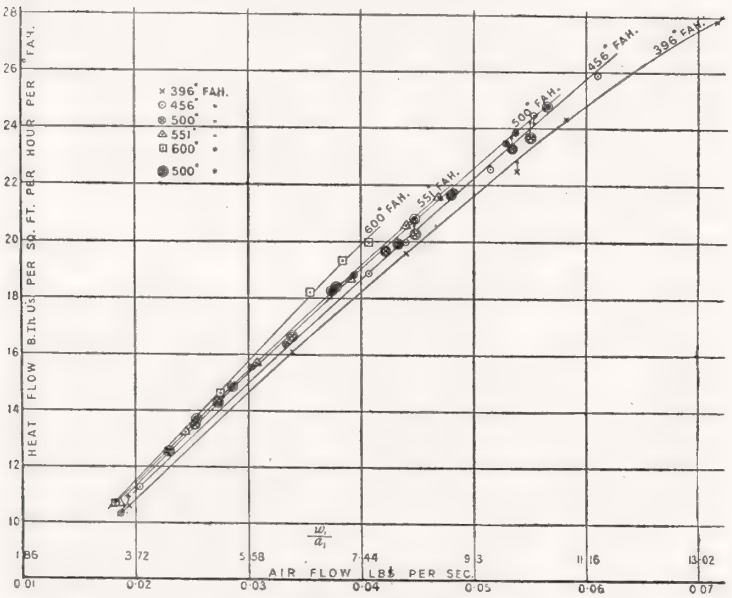


Fig. 3.—Heat Transmission per Sq.-ft. per Deg. Fah.

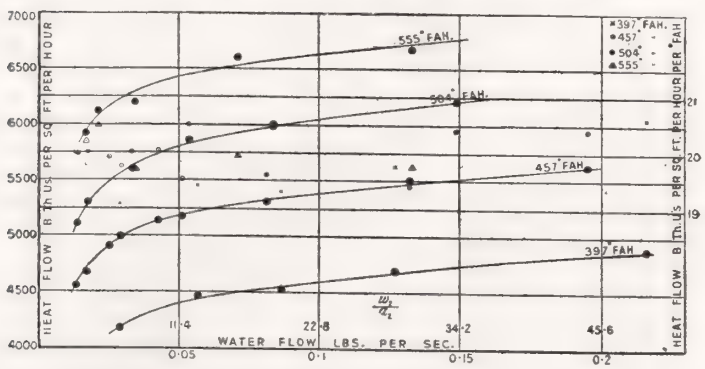


Fig. 4.—Tests with Different Rates of Water-Flow.

a_1 = sectional area of tube, in square feet.
w_1 = weight of gas-flow in lb. per sec. per tube.
c_1 and C_2 are coefficients.

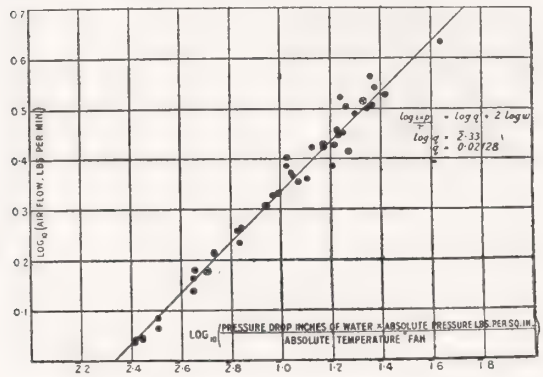


Fig. 5.—Tube Resistance to Air-Flow.

In applying this equation to economizer design all the values in the expression are known except the coefficients c_1 and C_2. The latter were determined by the experiments to be $c_1 = \cdot0011$ and $C_2 = \cdot013$. As an example of the application of the formula, the results given in Table I. were obtained on the justifiable assumption that c_1 and C_2 are practically constant over the conditions specified in this table. The coefficient of resistance to the flow of the gases through the tubes was also assumed to be practically constant. Taking a steam temperature of 377° Fahrenheit (199 lb. per square inch absolute), the gross saving in heat is, in this case, 9·6 per cent. Against this saving would have to be set the increased power and steam consumption of the extra fans required in connection with the system.

TABLE I.

Water per hour 24,000 lb.
Gases per hour 43,200 lb.
Boiler heating surface 3,000 sq. feet.

Gas temperatures: Inlet 500 Fah., outlet 268 Fah.
Water temperatures: Inlet 180 Fah., outlet 280 Fah.

$\frac{d_1}{}$ in.	d_2 in.	$\frac{d_2}{d_1}$	Gases per Tube lb. per Sec. w_1	$\frac{l}{}$ ft.	$\frac{l}{m_1}$	No. of Tubes.	Total Heating Surface sq. ft.	Ratio to Boiler H.S.	Heat per Sq. Ft. per Hour B.Th.U's.	Pressure Difference through Tube i, in. of Water.	Total Pressure Difference required, in. of Water.	Gases per Sq. Ft. Sectional Flow Area $\frac{w_1}{a_1}$
·75	1	1·33	·0125	6·9	441	960	1300	·423	1850	2·77	3·58	4·08
·75	1	1·33	·025	8·25	528	480	776	·259	3090	13·2	16·5	8·15
1·0	1·25	1·25	·0125	8·45	406	960	2120	·707	1130	·81	1·07	2·3
1·0	1·25	1·25	·025	9·6	455	480	1200	·4	2020	3·63	4·65	4·59
1·0	1·25	1·25	·05	11·8	566	240	740	·247	3240	18·1	22·3	9·18
1·25	1·5	1·2	·0125	10·1	388	960	3170	1·06	758	·32	·42	1·46
1·25	1·5	1·2	·025	11·0	422	480	1725	·575	1390	1·37	1·79	2·92
1·25	1·5	1·2	·05	12·9	495	240	1010	·338	2380	6·4	8·1	5·85

ECONOMY IN THE USE OF OIL AS FUEL FOR HARBOUR VESSELS.

By Engineer-in-Chief C. A. McALLISTER, U.S.N.

(*American Society of Naval Architects and Marine Engineers; November*, 1911.)

AFTER describing the oil fuel apparatus recently installed on board the harbour tug *Golden Gate*, doing revenue cutter boarding duty at San Francisco, California, the author proceeds to show the great economy effected thereby in comparison with the coal-burning system which it replaced. The *Golden Gate* is a vessel of the ordinary harbour tug type, 110ft. long overall, 20ft. 6in. beam, and 12ft. 0½in. moulded depth, the normal displacement being 220 tons. Her propelling machinery consists of a water-tube boiler and a vertical inverted triple-expansion engine having cylinders 13in., 21in., and 32¼in. diameter respectively, with a

stroke of 24in., and capable of developing a maximum I.H.P. of 550. The oil fuel apparatus, consisting of a horizontal cylindrical tank having a capacity of 23 barrels at 42 gallons, an oil pump $3\frac{1}{2}'' \times 2\frac{1}{4}'' \times 4''$, burners, etc., complete in all details, was fitted at a total cost of only $2,500. The fuel tank can be filled in ten minutes, and, as the tug's duties are very intermittent, carries sufficient oil for 10 days' ordinary use. For nearly nine-tenths of this time she must lie at the wharf with steam up ready for an immediate call, and it is during this period—when, if coal were used, she would be under banked fires—that a great saving in fuel is effected. The engineers' staff when coal was used for fuel consisted of four men: with oil it is found that three men can easily perform the work. The saving in the wages, food, and clothing of the fireman thus dispensed with amounts annually to $674. The average monthly cost of fuel for a nine months period, using oil, shows a saving of $173 per month compared with the cost of the coal used during an equal period before the change from coal to oil-burning was made, or at the rate of $2,075 per year. This added to $674, the saving of expense due to dispensing with one fireman, makes the total annual saving $2,749, which is a decidedly good return for an original investment of only $2,500. The contract price of coal in San Francisco during the period referred to was $5·40 per ton, and for oil $0·60 per barrel of 42 gallons. As the high cost of coal and the low cost of oil in San Francisco are probably the extremes which would be met with at any of the large American seaports, the saving at any other port would not be so marked, but would still, in the author's opinion, be sufficient in many harbours to warrant the necessary expenditure. For example, in New York, where the prices of the two kinds of fuel are not at such variance, an expenditure of $2,500 in the manner described would result in an annual dividend of over 46 per cent. on the investment.

Reciprocating Engines.

MARINE ENGINEERING DEVELOPMENT.

By Mr. E. Hall-Brown.

(*Presidential Address to the Institution of Engineers and Shipbuilders in Scotland ; October*, 1911.)

MR. Hall-Brown devoted his address to an endeavour to estimate at their proper value the various proposals at present being made to alter existing practice in marine engineering with the object of attaining greater efficiency and economy. From the time of the introduction of the tri-compound marine engine to the introduction of the marine steam turbine, no outstanding innovation was proposed and marine engineers became somewhat conservative in their ideas. The fact that the turbine, a novel invention entirely disregarding the traditions of marine engineering, has proved such an enormous success has, however, engendered a much more sympathetic attitude towards proposed developments in various directions. Nothing but good can come of the endeavour now being made to improve the economy of the marine engine ; but in order that this result may be obtained with a minimum expenditure, it is necessary that each possible line of advance should be carefully studied and its advantages and disadvantages accurately estimated.

As a starting point for such an investigation the author took an ordinary triple-compound engine working with saturated steam of 180 lb. boiler pressure, the steam being generated in ordinary return-tube boilers of the Scotch type. A well constructed engine of this class will have a mechanical efficiency of from 94 to 95 per cent. when driving its own air, circulating, and feed pumps, and can be designed to run at the speed at which the maximum propeller efficiency can be obtained. Such an engine will require 13½ lb. of steam per I.H.P. per hour, equivalent, with ordinary good coal burned in the furnaces of a Scotch boiler having a thermal efficiency of 70 per cent., to a consumption of 1·5 lb. per I.H.P. per hour. Built, as it usually is, with three cylinders and three cranks placed at angles of 120 degrees, such an engine gives a fairly uniform turning moment, and the loads upon the various parts are moderate in relation to the working pressure. It is, therefore, a fairly cheap engine to build and to keep in repair. An additional advantage, and that of great importance, is that the management of such an engine presents no difficulties to the average sea-going engineer.

There are possibilities of attaining greater economy in the use of fuel without departing from the reciprocating engine. Without any great alteration to the engine, superheated steam may be safely used in an ordinary triple-compound engine. The saving which may be obtained thereby is very considerable, and may amount to 15 or 20 per cent. of the coal consumption with a superheat of from 150° to 170° F. An interesting attempt to attain a high degree of fuel economy by the use of extremely high working pressure in conjunction with superheating, steam jacketing, and multi-stage feed-heating is to be found in the five-crank quadruple-expansion engines of the steamers *Inchdune* and *Inchmarlo*, built at the Central Marine Engine Works, West Hartlepool. The working pressure was 265 lb. per square inch, and the degree of superheat about 60° F. The consumption of coal on actual service is said not to exceed 1 lb. per I.H.P. per hour for propulsive purposes.

The condensation of steam and the production and use of high vacua have recently received much consideration ; and there is reason to believe that a very high vacuum, which has always been recognised as necessary for maximum economy in the steam turbine, is also conducive to economy in a reciprocating engine.

Economizers for heating the boiler feed water by means of the waste gases have for very many years formed part of every land installation having any claim to efficiency. For marine work, Mr. Hall-Brown is of opinion that good results would be obtained from an economizer consisting of small tubes, so as to ensure a high speed of water over the heating surface, together with a fan to induce sufficient draught, somewhat on the lines advocated by Professor Nicolson in his paper on " Boiler Economics " read before the Institution last session. Such an apparatus would probably add from 8 to 10 per cent. to the net efficiency of the boiler installation after allowing for the power necessary to drive the fan.

To show what is actually being accomplished

Photo by] [*Warneuke, Glasgow.*

Mr. E. HALL-BROWN,
President of the Institution of Engineers and Shipbuilders in Scotland.

Yours faithfully
E. Hall-Brown

in land boiler installations to-day, the author gives in an appendix some particulars of tests made with steam-generating plant of Messrs. Babcock & Wilcox's construction, in one of which the combined efficiency of boiler, superheater, and economizer is given as 87·9 per cent.

Mechanical stoking as a source of fuel economy is another point worthy of serious consideration.

In connection with the use of fuel, a most interesting development has been inaugurated by Professor Bone, of Leeds University, by the combustion of gaseous fuel in a special type of boiler upon what is known as the process of "surface combustion." With an experimental boiler and a very simple form of economizer Professor Bone has obtained the remarkable efficiency of 94·3 per cent., but there would in most cases be a deduction from this figure for the power necessary to supply the gas and air under pressure.

The results which have been attained with reciprocating engines on land are sufficient to show that there is considerable possibility of improvement in marine engine design if the subject is approached with an open mind. Thus the test of a Sulzer engine at Messrs. Singer's works shows that, with a reciprocating engine of the triple-compound type capable of developing 1,600 I.H.P., the steam consumption need not exceed 8·76 lb. per I.H.P.-hour. A very different type of engine—the vertical high-speed engine constructed by Messrs. Belliss & Morcom, Ltd.—has shown on test a steam consumption of 10½ to 11 lb. per B.H.P.-hour, with a superheat of slightly over 200° F. and a steam pressure of 180 lb. Such an engine, with a speed-reducing gear, would make a very good showing when compared with a turbine set of the size fitted to the s.s. *Vespasian*. Another very simple engine as regards the number and construction of its working parts—the locomobile, by Messrs. Wolf—has under tests given a coal consumption less than 1 lb. per B.H.P.-hour.

The introduction of the marine steam turbine did not in its earlier stages seriously affect the supremacy of the reciprocating engine, but at the present time this state of affairs has changed; and although it is not clear that the balance, as regards fuel economy, inclines more to one type than to the other, there are directions in which the turbine is undoubtedly supreme; and the recent successful introduction of speed-reducing gear has further widened its field of application.

The most economical speed of a steam turbine is much too high for the screw propeller, more especially in the case of vessels of moderate speed, and it was in this connection that the demand for a speed-reducing gear first became acute. There are at least three principal systems of speed-reducing gear proposed, *viz.*, machine-cut toothed gears as fitted in the s.s. *Vespasian*; electric transmission from a dynamo driven by the steam turbine to an electric motor on the propeller shafting; and the hydraulic system associated with the name of Föttinger. The high efficiency of accurately-cut toothed gears and the success which has attended the installation in the *Vespasian* will have the tendency to incline mechanical engineers to favour that system.

There is no doubt that the thermal efficiency of the steam turbine, as used for land purposes, is in excess of that of any other type of steam engine. The tests of a turbo-generator of 6,000 K.W. capacity, particulars of which have been supplied to the author by Messrs. Richardsons, Westgarth and Co., Hartlepool, show that in this case the steam consumption was 8½ lb. per B.H.P.-hour, the working pressure being 190 lb. per square inch, and the superheat 176° F. Such an engine, receiving steam from a steam-generating plant having a thermal efficiency of 85 per cent., would have a coal consumption at the rate of practically ·9 lb. of coal per B.H.P.-hour, equivalent to somewhat less than ⅞-lb. of coal per I.H.P.-hour in a reciprocating engine.

Various proposals for combined reciprocating and turbine sets have been made and carried into effect. Designed to secure at once the efficiency of the reciprocating engine at the high-pressure end and that of the turbine at the low-pressure end, the combination has probably failed to take full advantage of the theoretical saving possible with the system, because of the limitations imposed by the propeller and the loss inevitably arising from the use of an additional screw and line of shafting.

At the present time the attention of the marine engineering world is being concentrated upon the development of the internal-combustion engine for marine propulsion. Such engines may be constructed for the use of either gas or oil fuel, but the fact that a gas producer is required to prepare suitable fuel gas complicates the problem so far as the gas engine is concerned, and the movement to-day is entirely in the direction of using oil as fuel. With the exception of such engines as are being made for smaller vessels, attention is in the meantime concentrated upon engines working on the Diesel cycle. The chief difference between a Diesel engine and an internal-combustion engine of any other type at present being manufactured results from the degree of compression used, which in the case of the Diesel engine raises the temperature of the compressed air to a point sufficient to ignite very heavy oils when properly pulverized or atomized during the period of

injection into the cylinder. In this way no ignition apparatus is required, and the ignition is timed by the injection of the fuel. The method of injection is also such as to produce a period of actual combustion within the cylinder, during which period the initial pressure of 500 lb. per square inch is approximately maintained. For marine purposes the Diesel engine is being made on two different cycles, the four-stroke cycle and the two-stroke cycle. The former has the higher mechanical efficiency, and will probably be more economical of fuel than the latter; but the fact that the two-stroke cycle engine has two working strokes for every working stroke of the four-stroke cycle engine, will probably turn the scale in favour of the two-stroke cycle motor for large engines.

That marine engineers should calmly contemplate the use of an internal-combustion engine having, for the two-stroke cycle, at least three double-acting cylinders or six single-acting cylinders, each having on each working end a scavenging valve, a fuel valve, and a compressed-air starting valve, each with its cam, roller, lever, and springs, together with the pumps for the scavenging air, and a multiple stage air compressor for starting, manœuvring and spraying the fuel, shows the enormous change in the mental outlook of the marine engineering world. One is inclined to wonder if equally drastic proposals regarding the construction of the steam engine would receive equally favourable consideration.

It is usually stated that the consumption of fuel oil in a Diesel engine is ·4 lb. per B.H.P.-hour. It would seem that it is possible, by care in design and attention to every source of economy, to reduce the coal consumption of a steam-driven vessel to 1 lb. per I.H.P.-hour. If suitable coal could be bought at 12s. per ton, a Diesel-engined vessel would require to obtain oil fuel at about 32s. per ton to be equally economical as regards the cost of fuel. Comparisons of economy should also take into account the fact that the amount of fuel to be carried by the Diesel-engined ship will be much less than that of the coal-driven vessel, and therefore in a vessel of given size more cargo deadweight can be carried. On the other hand, the first cost of a Diesel engine exceeds that of steam plant.

In conclusion, Mr. Hall-Brown said he was a firm believer in the future of the internal-combustion engine, both on sea and land. He was not, however, convinced that the final development will be along the present lines. As regards reliability, ease in adjustment, and simplicity in manipulation, the marine steam engine is yet without a rival. If to these advantages be added increased economy, such as he believed possible of attainment, it is doubtful if any internal-combustion engine yet proposed will take its place in ordinary merchant vessels.

RELATIVE POSSIBILITIES OF THE DIESEL OIL ENGINE, GEARED TURBINE, AND SUCTION GAS ENGINE, AS COMPARED WITH THE RECIPROCATING ENGINE, FOR MARINE PROPULSION.

(*North-East Coast Institution of Engineers and Shipbuilders; April*, 1912.)

At the suggestion of the Council of the Institution, the authors dealt more particularly with the propelling machinery suitable for an ordinary cargo vessel of medium size and moderate speed.

THE DIESEL OIL ENGINE.

By Mr. E. L. Orde.

IT appears to the author that the extent to which the earning power of a cargo vessel is affected by the adoption of the Diesel engine has not so far received the consideration at the hands of engineers and shipbuilders which it deserves. To open the question the author submits two outline plans, Fig. 1 showing a shelter-deck cargo steamer with reciprocating engines, and Fig. 2 the same vessel re-arranged and fitted with Diesel engines. The general particulars of the two vessels are given in Table I. To obtain an estimate of the cost of working the two vessels, a distance of 3,500 knots is taken to represent a fair average voyage, and the bunkers in both vessels are arranged accordingly. The cost of transporting one ton of cargo for this distance is taken as the unit cost of working, and is shown in Fig. 3 for the two vessels with fuel at prices ranging from 10s. to 50s. per ton, an example of the calculations made in order to construct this diagram being given in Table II. The comparison shows, for example, that with coal at 15s. per ton the cost of oil in the Diesel ship must not exceed 54s. per ton as the economical limit within which the Diesel ship shows to advantage.

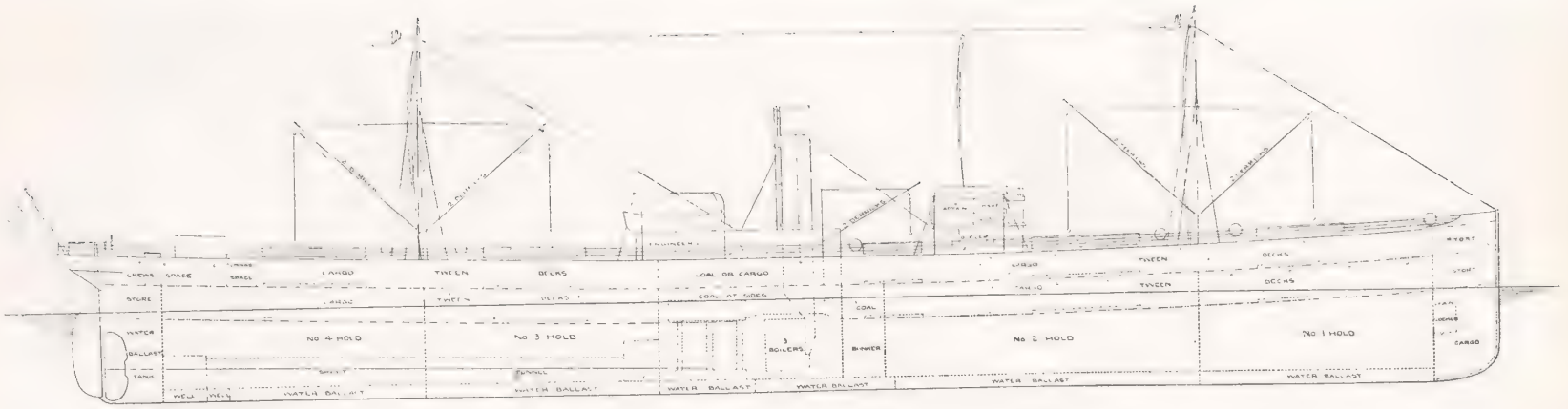


Fig. 1—Shelter-deck Cargo Vessel with Reciprocating Engines.

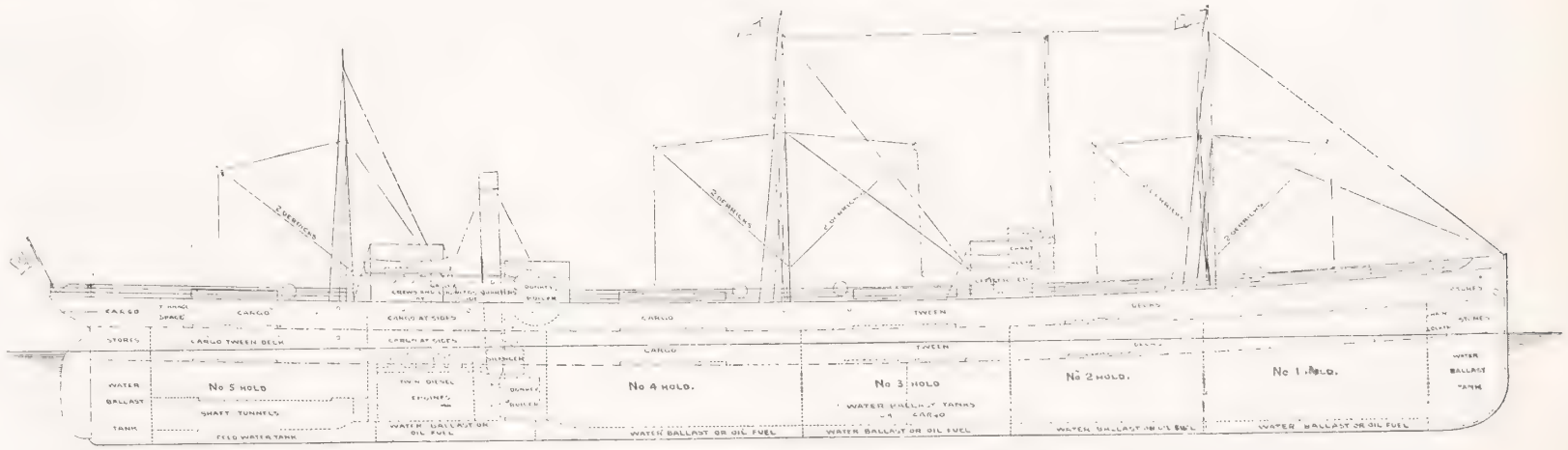


Fig. 2.—Shelter-deck Cargo Vessel with Diesel Engines.

The propelling machinery in the steamship is of the usual three-crank triple-expansion type, with cylinders 27, 45 and 75 by 48in., and three boilers 14ft. 6in. by 11ft. 6in., fitted with Howden's forced draught. The actual results from a number of different voyages show an average of 2,400 I.H.P. and a consumption of 1·6 lb. per H.P. per hour for all purposes.

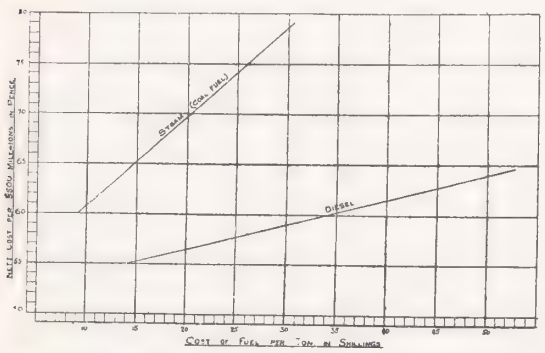


Fig. 3.—Comparative Cost of Working Steam and Diesel-propelled Vessels.

The propelling machinery for the Diesel ship is of the single-acting two-stroke cycle type, designed to develop 2,150 B.H.P. in ordinary service at sea at about 115 revolutions. The two-stroke cycle engine has been adopted for this comparison, as the balance of advantages seems to be with it rather than with the four-stroke cycle engine. The weight of the two-stroke engine is about 12 to 15 per cent. and the cost about 10 to 12 per cent. less than that of the four-stroke cycle engine. On the other hand, the latter has a considerably lower fuel consumption —according to some authorities equal to 10 per cent. The influence of cost and weight of machinery on the earning power of cargo vessels is, however, considerable, and a point is soon reached when the saving due to the lower consumption of the four-stroke cycle engine is counterbalanced by the capital charges on the higher cost and the loss on freight-earning cargo shut out by the extra weight of the engine. In the vessels under consideration, for a 3,500-knot voyage this critical point appears to occur when the cost of the oil is 20s. per ton.

With regard to the question of driving the auxiliary machinery of the Diesel ship, the author is doubtful about the advantages of electric power or compressed air for this purpose, and in the type ship steam power is proposed, the steam being supplied by two oil-fired boilers, one for use at sea and the other in port.

Table I.—Shelter Deck Cargo Vessel.

	Feet.	Inches.
Dimensions:		
Length overall	412	0
Length between perpendiculars	400	0
Breadth	52	0
Depth	29	9

	Steam Engines.	Diesel Engines.
Gross tonnage	4,655	4,800
Net tonnage	2,930	3,030
Draught	26′ 1″	26′ 1″
Total deadweight in tons	8,640	8,775
Average sea power	2,400 I.H.P.	2,150 B.H.P.
Average sea speed in knots	10½	10½
Radius of action in knots	3,500	3,500
Fuel consumption—tons per day	42	12·7
	(=2,400 I.H.P. @ 1·6 lb. per I.H.P. hour.)	(2,150 @ ·55 lb. per B.H.P. per hour.)
Freight-earning cargo in tons	7.880	8,530
Freight-earning cargo per ton, total D.W.	·914	·972
Freight-earning cargo per ton, gross register	1·69	1·77
Freight-earning cargo per ton, net register	2·68	2·82
Capacity for cargo (grain) in cubic feet	488,000	523.800
Cargo capacity per ton of freight-earning cargo in cubic feet	62	61·5

TABLE II.—APPROXIMATE ESTIMATE OF THE COMPARATIVE COSTS OF WORKING STEAM AND OIL-PROPELLED CARGO VESSELS.

	OIL ENGINE.		STEAM ENGINE.	
CAPITAL.	£78,000.		£63,000.	
	Per Voyage.	Per Month.	Per Voyage.	Per Month.
Insurance	—	490	—	370
Fuel (oil @ 40/-, coal @ 15/-)	360	—	445	—
Wages and provisions	—	350	—	410
Wear and tear	—	110	—	100
Deck and engine room stores	—	110	—	100
Port charges @ 5/- per ton	755	—	735	—
	1,115	1,060	1,180	980
16 voyages	17,840		18,880	
12 months	12,720		11,760	
	30,560		30,640	
5% depreciation	3,900		3,150	
Management	500		500	
	£34,960		£34,290	
Tons freight-earning cargo carried	$16 \times 8,530 = 136,480$		$16 \times 7,880 = 126,080$	
	$34,960 \times 240 = 8,390,400$d.		$34,290 \times 240 = 8,229,600$d.	
	136,480 = 61·4d.		126,080 = 65·2d.	

GEARED TURBINES.

By the Hon. Sir CHARLES A. PARSONS, K.C.B., F.R.S., and Mr. R. J. WALKER.

THE problem of applying the steam turbine to vessels of moderate and slow speeds may be said to have been satisfactorily solved by the association of mechanical gearing with high-speed turbines. The principal advantages of geared turbines as compared with triple-expansion engines are summarized by the authors as follow :—Increased economy in coal and oil consumption ; absence of racing in rough weather ; reduced wear and tear of engines ; reduced stress on screw shafting, due to even turning moment of engines and absence of racing ; reliability of running in all weather ; reduced weight of machinery ; increased dead-weight cargo, due to increased economy in coal consumption and reduced weight of engines.

With geared turbines a consumption of saturated steam of about 12 to 13 lb. per S.H.P. for the main engine can be obtained, which is equivalent to about 11 to 12 lb. per I.H.P. for reciprocating engines, assuming a ratio of S.H.P. to I.H.P. of 90 to 92 per cent. Further advantages can be secured by the adoption of superheated steam and by burning oil fuel in the boilers.

Tables III. and IV. have been prepared to show

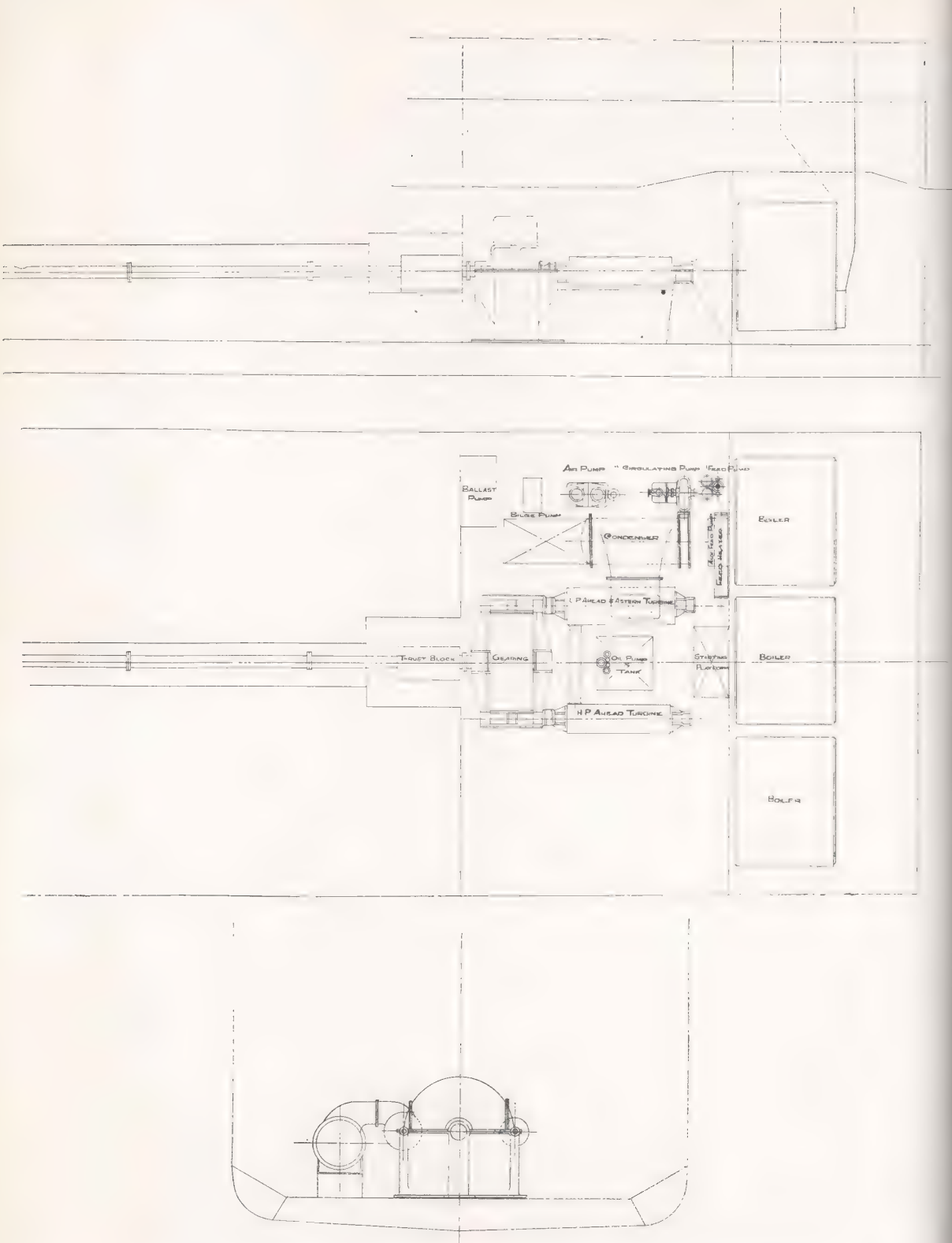


Fig. 4. - Arrangement of Geared Turbines for Single-Screw Cargo Steamer.

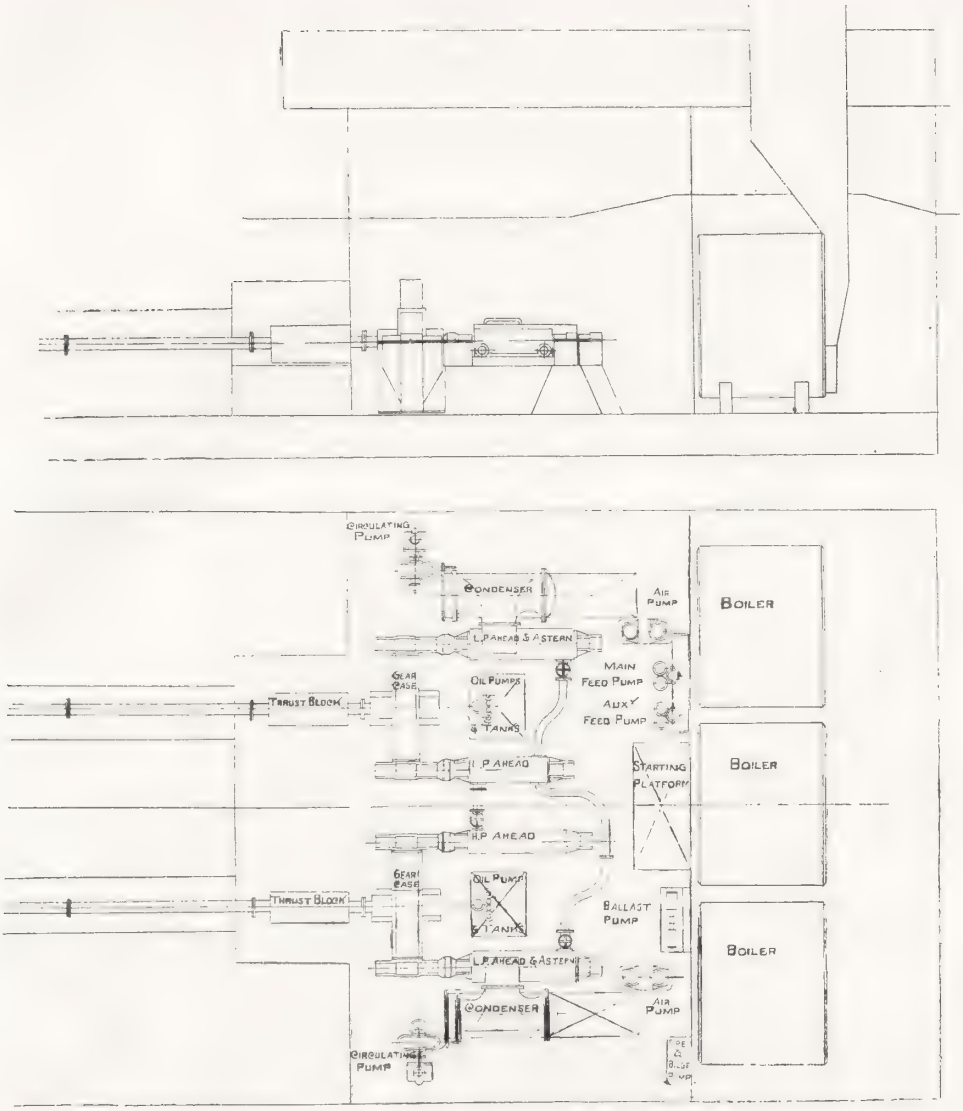


Fig. 5.—Geared Turbines for Twin-screw Cargo Steamer.

the possibilities of geared turbines when compared with triple-expansion engines in the case of a vessel of the dimensions given. Column *A* refers to the vessel with reciprocating engines; column *B* refers to a proposal for geared turbines on a single shaft, as shown in Fig. 4; and column *C* to geared turbines with twin screws, as shown in Fig. 5.

The machinery in Fig. 4 consists of two turbines in "series," *viz.*, one high-pressure and one low-pressure turbine. The reversing turbine is incorporated in the exhaust casing of the low-pressure turbine. The diameter of the gear-wheel is about 12ft., and of the pinions 6·3in. The total width of the face of the wheel is 32in. The inclination of teeth is 45° to the axis—double helical. The revolutions of the propeller would be 66, and of the turbine about 1,500.

The machinery shown in Fig. 5 consists of one high-pressure, one intermediate-pressure, and two low-pressure turbines. Steam is admitted to the h.p. turbine on one side of the vessel, exhausts into the i.p. turbine on the opposite side of the vessel, and thence passes to each l.p. turbine. A reversing turbine is fitted in the exhaust casing of each l.p. turbine. The diameter of each gear wheel would be about 10ft. 6in., and the pinions 5in.; the width of wheel face would be 15in., and the revolutions of propeller 110 and of the turbines about 2,800.

The authors estimate that, apart from the question of superheat and oil fuel, the saving of

TABLE III.

TYPE OF PROPELLING MACHINE.	A. Triple Expansion Engines.	B. Geared Turbine.	C. Geared Turbine.
Number of propellers	1	1	2
Revolutions	66	66	110
Diameter of propeller	18 feet	18 feet	12 feet
Pitch of propeller	18 feet	18 feet	12 feet
Length of vessel B.P.	400 feet	400 feet	400 feet
Breadth extreme	52 feet	52 feet	52 feet
Depth moulded	29 ft. 9 in.	29 ft. 9 in.	29 ft. 9 in.
Deadweight	8,465 tons	8,535 tons	8,540 tons
Draft of water	26 ft. 1 in.	26 ft. 1 in.	26 ft. 1 in.
Displacement	11,760 tons	11,760 tons	11,760 tons
Speed on service	10½ knots	10½ knots	10½ knots
		Equivalent	Equivalent
I.H.P.	2,400	2,400	2,400
S.H.P. assuming ratio S.H.P. to I.H.P. of 90 per cent.	—	2,160	2,160
Weight of propelling machinery:—			
Propeller shafting, thrust block and fittings aft of engine room after bulkhead	64 tons	64 tons	62 tons
Engines, condensing plant, auxiliary machinery and all fittings within engine room	180 tons	146 tons	143 tons
Boilers, fans, funnels and all fittings within boiler rooms	281 tons	245 tons	245 tons
Total steaming weight	525 tons	455 tons	450 tons

Photo by The Hon. Sir CHARLES A. PARSONS, K.C.B., F.R.S., &c. [*Elliott & Fry, London.*

Charles A. Parsons

15 per cent. in coal consumption which can be effected by the adoption of geared turbines instead of reciprocating engines means, in the case of a vessel of the dimensions and power stated, a gain to the shipowner in the neighbourhood of £2,000 per annum.

With regard to actual experience with geared turbines at sea, the authors state that since the *Vespasian* was first put on service with her new propelling machinery she has completed some 62 voyages, covering a distance of over 37,000 knots. During this period the gearing has not given the least trouble, nor has any appreciable wear been detected on the teeth of the pinions or gear wheels.

Table IV.

	A.	B.	C.
Estimated consumption saturated steam main engines per S.H.P.	—	11·8 lb.	11·25 lb.
Estimated steam consumption auxiliaries per S.H.P. of main engines	—	1·2	1·2
Estimated total consumption of saturated steam per S.H.P., all purposes	—	13·0	12·45
Estimated total consumption of saturated steam per I.H.P., all purposes	15·5 lb.	Equivalent 11·7	Equivalent 11·2
Estimated coal consumption per I.H.P., all purposes	1·55	Equivalent 1·17	Equivalent 1·12
Estimated coal consumption per S.H.P., all purposes	—	1·3	1·25
Estimated consumption of steam (100° superheat) per S.H.P. main engines	—	10·6	10·2
Estimated consumption auxiliaries per S.H.P. of main engines	—	1·2	1·2
Estimated consumption of steam (100° superheat) for all purposes, per S.H.P. of main engines	—	11·8	11·4
Estimated steam consumption per I.H.P. of main engines (with 100° superheat), all purposes	—	Equivalent 10·6	Equivalent 10·2
Estimated consumption of oil per I.H.P. (saturated steam), all purposes	—	Equivalent ·81	Equivalent ·78
Estimated consumption of oil per S.H.P. (saturated steam), all purposes	—	·90	·86
With oil fuel and 100° superheat—estimated oil consumption, all purposes, per I.H.P. of main engines	—	Equivalent ·76	Equivalent ·73
With oil fuel and 100° superheat—estimated oil consumption, all purposes, per S.H.P. of main engines	—	·85	·81

For a vessel of the size under discussion Mr. Holzapfel proposes to divide the propelling machinery into two units driving twin screws. The producer plant would be capable of supplying gas for 3,000 H.P., and would consist of four producers, each 5ft. 6in. square by 15ft. high, placed in a recess 15ft. by 15ft. amidships abaft the fore engine-room bulkhead, and completely enclosed by gas-tight bulkheads. The cooling towers would be placed on a platform just above the water level, and would be four in number, about 7ft. diameter and 20ft. high. From the coolers the gas would pass to two 25-H.P. electrically-driven centrifugal tar extractors, and from thence, under slight pressure induced by

THE SUCTION GAS ENGINE.

By Mr. A. C. Holzapfel.

MR. Holzapfel maintains that a suction gas plant, working in conjunction with Föttinger transformers, will ultimately be the cheapest form of propelling machinery in first cost, and will have the advantage of using a cheap quality of coal which can be bought on the Tyne at from 8s. to 9s. per ton, instead of oil, which must always be dear in a country having no supplies of its own.

the fans of the tar extractors, to two dry scrubbers before passing to the engines.

The prime movers would consist of two six-crank tandem gas engines, each of about 1,400 B.H.P. Each engine would have 12 cylinders of about 20in. diameter, with a stroke of 18in., which is small enough to make piston cooling unnecessary. At 300 revolutions per minute the piston speed would be 900ft. per minute. As regards ignition, either high and low-tension Bosch magneto, or else low-tension Bosch magneto and Lodge accumulator ignition would be used. The main engines would be started electrically, and not by compressed air. The auxiliary machinery would be electrically driven.

The gas engines not being reversible, a Föttinger transformer would be fitted to each engine, so adjusted as to reduce the revolutions from 300 to 100 on the propeller shaft. The transformer would absorb about 12 per cent. of the power developed by the engines. The fuel consumption would be about ·9 lb. of bituminous coal per B.H.P.-hour, which, for 2,800 B.H.P., amounts to 26½ tons per day of 24 hours. This figure covers the consumption due to all auxiliary machinery.

The author estimates the weight of the gas plant, including cooling water in the tanks of the cooling towers and fuel in the producers, at about 220 tons; that of the two gas engines at 120 tons; two transformers, including water, at 60 tons; two oil engines and dynamos at 25 tons; and pumps, electrical motors, gas pipes, coolers, etc., complete at a further 50 tons; making a total weight of 475 tons.

As regards the cost, the two gas engines would cost about £7,500; two transformers, £4,000; gas plant complete, about £2,000; oil engines, dynamos, steering gear, electric winches, and windlass, about £3,200; total about £16,700.

In conclusion, the author admits that the gas engine and plant in their present form are capable of vast improvement as regards marine work, but holds that the possibilities of this type of prime mover are such as to deserve the careful consideration of all marine engineers.

THE NAVAL RECIPROCATING ENGINE: ITS CHARACTERISTICS, DIMENSIONS, AND ECONOMICS.

By Mr. E. N. Janson.

(*American Society of Naval Engineers; February*, 1912.)

THE author deals in a comprehensive way with the design of reciprocating engines as developed in connection with propelling machinery of this type installed in American warships. In the course of the first part of the paper, which treats upon factors influencing design, the author states that the present engine used for battleship installations is in the nature of a standard type. The initial steam pressure at the valve chest in the latest battleship engines is taken at 265 lb. gauge, with a pressure at the boilers of 295 lb. gauge. The piston speed used is about 1,050 feet per minute for speeds of from 20 to 20½ knots. The engines for all the high-speed battleships are made with 48in. stroke, the revolutions, according to speed, varying between 115 and 130 per minute at full power.

The average back pressure in naval engines is about 6 lb. absolute. This pressure, however, in the *Delaware*, when running at full power and 128 r.p.m. with about 26·3in. of vacuum, was only 4½ lb., and at 87 r.p.m. did not exceed 1½ lb.

Superheated steam has been used in some of the later ships. The *South Carolina* and *Michigan*, the *Delaware* and the *North Dakota*, had superheaters fitted in all boilers, giving an average of about 75° F. superheat. In the two latest ships, the *Oklahama* and *Nevada*, the fitting of superheaters is not required.

The type of engine generally used in battleship installations of the United States is the four-cylinder triple-expansion engine illustrated in Fig. 1. The distribution of power among the cylinders is such as to give about one-third the total to each of the high-pressure and intermediate-pressure cylinders, and about one-sixth to each of the low-pressure cylinders.

The author then deals with sources of loss in the engine, factors conducive to economy, the determination of cylinder diameters, and the question of steam consumption, and gives an interesting statement of the actual water rate in lb. per I.H.P. per hour as measured on the trials of various vessels, which is reproduced in Table I.

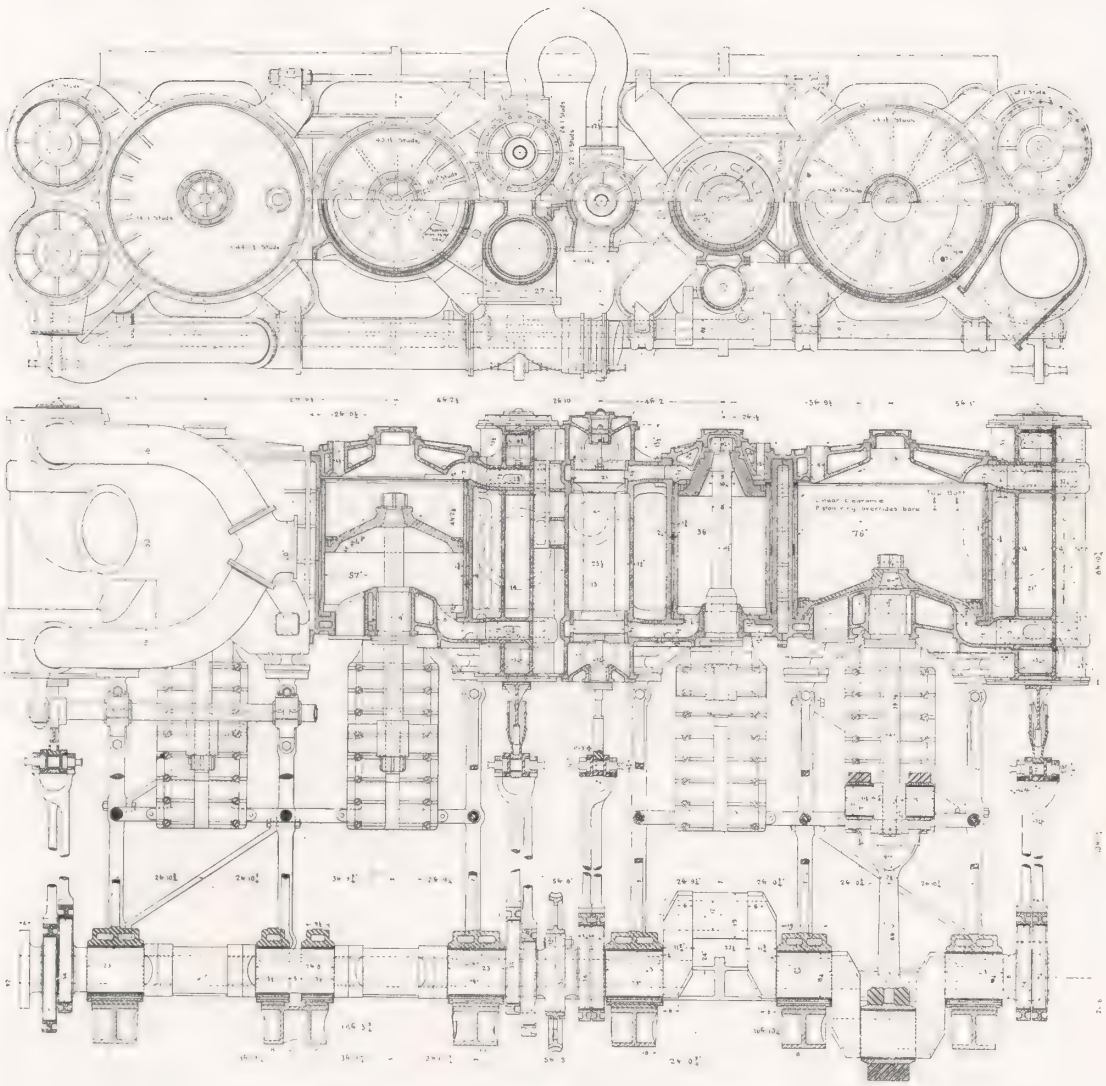


Fig. 1.—Starboard Engine, U.S.S. "Delaware."

TABLE I.

ACTUAL WATER RATE IN LB. PER I.H.P. PER HOUR AS MEASURED DURING TESTS OR ON TRIAL.

Name of Ship.	Percentage of Power.	Steam at Engine.		Vacuum, inches in Condenser.	Cylinder Ratio.	Piston Speed in ft.	Water in lb. per I.H.P. hour, measured.	Superheat.	Characteristics of Engine.
		Pressure by Gauge.	Temperature.						
			°F.					°F.	
S.S. *Iona*	86	165	373	28·2	6·71	391	13·35	...	One 3-cyl. triple-expan. engine; H.P. cyl. steam jacktd.; all slide valves.
S.S. *Zenith City*	Full power	192	384	22·85	8·2	610	15·42	...	One 3-cyl. triple-expan. eng.; no steam jackets; slide valves on I.P. and L.P.
H.M.S. *Hermes*	Full power	172	377	.....	.....	.....	15·64	..	Two 4-cyl. triple-expan. engines.
S.S. *Pennsylvania*	10	229	393	.....	.....	.....	21·9	...	One cyl. quadr.-expan. engine; slide valve on L.P. cyl.; cyls. all unjacketed.
	91	237	401	22·73	12·63	530	14·4	...	
	18	.....	.....	21·16	12·63	300	20·2	...	
U.S.S. *Birmingham*	Full power	234	395	26·5	9·6	1,150	17·4	...	Two 4-cyl. triple-expan. engines; cyls. all unjacketed; piston valves all cyls.
	67	187	382	27·0	9·6	1,032	16·6	...	
	10	201	388	26·8	9·6	547	20·2	...	
U.S.S. *Delaware*	Full power	285	478	26·3	7·8	1,027	13·38	61	Two 4-cyl. triple-expan. engines; barrels of all cyls. and top and bottom of I.P. and L.P. steam jacketed.
	58·8	272	463	27·16	7·8	868	12·85	50	
	13·6	252	443	27·97	7·8	544	15·12	36	
Yacht *Idalia*	Full power	190	383	25·5	8·0	582	18·3	0	One 4-cyl. triple-expansion engine.
	Do.	196	443	25·9	8·0	574	17·0	57	
	Do.	198	483	25·4	8·0	574	15·8	96	
	Do.	203	494	25·2	8·0	580	15·5	105	

THE BEST ARRANGEMENT FOR COMBINED RECIPROCATING AND TURBINE ENGINES OF STEAMSHIPS.

By Mr. G. W. DICKIE.

(*American Society of Naval Architects and Marine Engineers; November*, 1911.)

THE arrangement which has found favour with those responsible for the most important "combined" installations on board ship has been that of two reciprocating engines and one turbine, the steam being expanded in the reciprocating engine down to a pressure of 9 to 10 lb. absolute. Mr. Dickie is of opinion that with installations up to 20,000 H.P. better results could be secured by having one reciprocating engine on the centre line of the ship, delivering steam to a turbine on each side at about 30 lb. absolute, and arranged as outlined in Fig. 1. This pressure is slightly below that at which steam is usually delivered to the low-pressure turbine in a three-turbine set, and would admit of the reciprocating engine developing about 40 per cent. of the total power. The side propellers could be run at a higher number of revolutions than would be advisable for a centre turbine-driven propeller, so that the side turbine rotors and casings need only be of moderate diameter. In a 20,000 H.P. set, with the centre

shaft making, say, 78 revolutions, the author would have the side shafts running about 400 revolutions, the necessary diameter of the rotor being about 8 feet.

With the reciprocating engine closing its work with a terminal pressure of 30 lb. absolute, a simple compound engine should give satisfactory results. Where the total power was to be above 8,000 the author would make the reciprocating engine with two high and two low-pressure cylinders. Both pairs of cylinders would have piston valves. By a proper arrangement of exhaust from the low-pressure cylinders only one change valve would be necessary.

Objection may be taken to the proposed arrangement on account of the reduced manœuvring power, due to having only one propeller capable of reversal instead of two, and also on account of the total loss of backing power in the event of a break-down of the reciprocating engine. With regard to these objections, Mr. Dickie considers that the advantage of a simpler and cheaper arrangement with the most effective propeller on the centre line of the ship, and the promise of better economy, should outweigh any possible advantage for manœuvring which the usual arrangement may possess, an advantage possibly overrated. The second objection is admitted, but it is considered that with the simple construction of the proposed reciprocating engine a break-down involving complete stoppage seems rather a remote possibility.

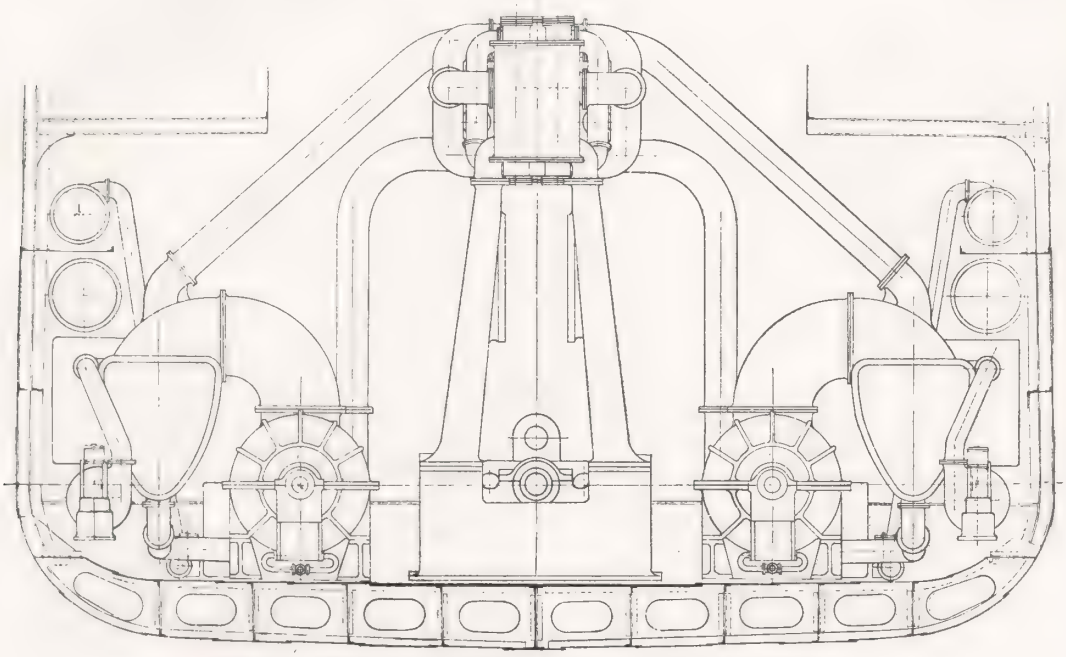


Fig. 1.—Proposed Arrangement of Combined Engines.

Steam Turbines.

THE PARSONS MARINE STEAM TURBINE, AND ITS APPLICATION TO VARIOUS CLASSES OF VESSELS.

By Mr. E. F. B. Anderson.

(*American Society of Naval Architects and Marine Engineers; November*, 1911.)

THE author describes some of the various arrangements of propelling machinery in past and present vessels which are fitted with Parsons turbines. In torpedo boats and small fast steam yachts a similar arrangement of machinery is still installed to that adopted in the pioneer vessel *Turbinia*. In destroyers the standard three-shaft arrangement is usual, and includes two cruising turbines in series, arranged at the forward end of each low-pressure turbine. Among the smaller cruisers and scout cruisers a four-shaft arrangement is largely adopted. In battleships and cruisers of the *Dreadnought* type four shafts are fitted, which are driven by two independent sets of ahead turbines, consisting of two high-pressure on the outboard shafts and two low-pressure on the inboard shafts. Two cruising turbines are arranged in parallel at the

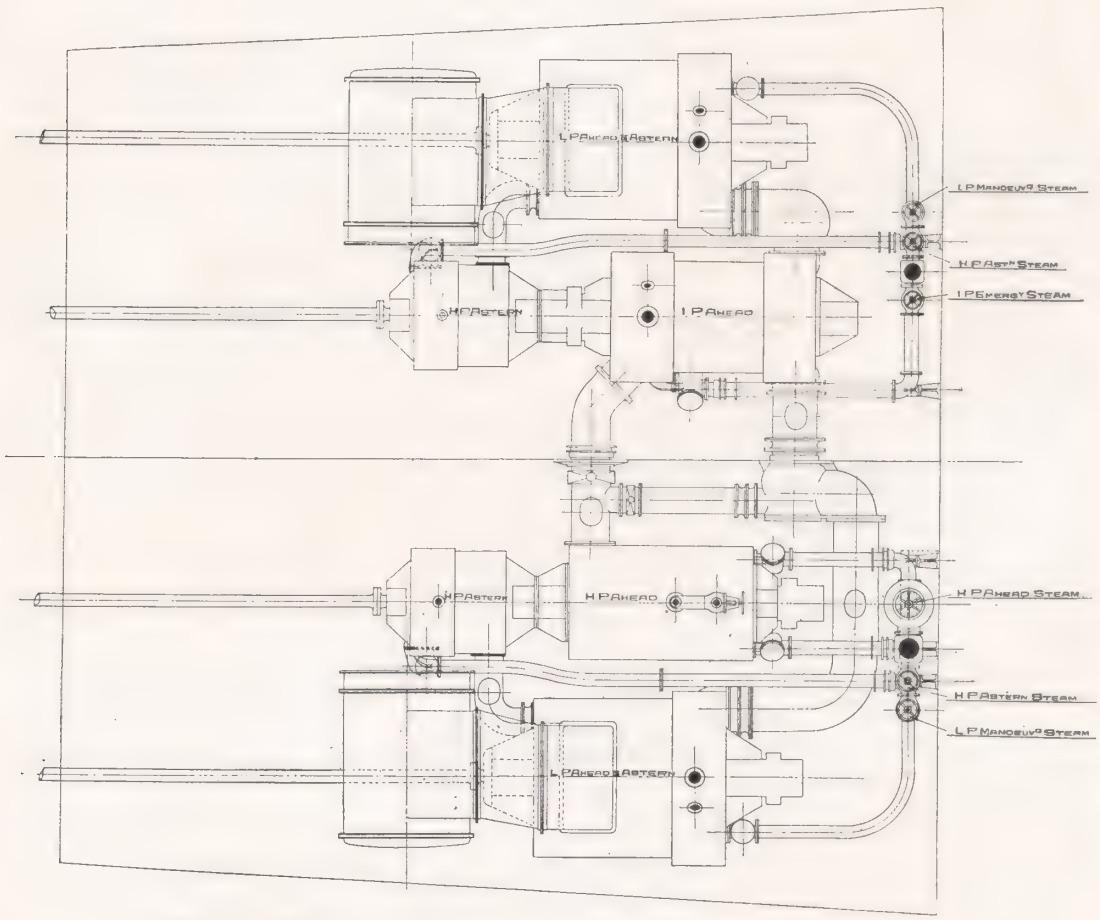


Fig. 1.—General Arrangement of Turbine Machinery in Series.

forward end of each low-pressure turbine. All shafts are arranged with astern turbines. In the latest ships of this type cruising turbines have been dispensed with. Battleships building for all the leading Powers are being installed with Parsons turbines. The main difference in the machinery arrangement of the French and United States vessels compared with those of the *Dreadnought* type is that the cruising turbines are arranged in series instead of in parallel. Some of the German vessels have four shafts with the turbines arranged in series, while three battleships are being built having triple screws and Parsons turbines, but with details special to the German Navy. In Spain three battleships are in course of construction, which will have the turbine machinery arranged as in Fig. 1. No cruising turbines are to be fitted, but one high-pressure and one intermediate-pressure on separate shafts take the place of the usual two high-pressure turbines, and the intermediate-pressure is designed to divide its exhaust steam into two low-pressure turbines arranged on the outboard shafts. In the mercantile marine, the three-shaft arrangement of turbine machinery similar to that installed in the first ship, the *King Edward*, is still used in almost all ships except those of very large power. The Cunard liners *Lusitania* and *Mauretania* have four-shaft arrangements of turbines, which consist of two independent sets. A modified arrangement of turbines has been adopted in the new French liner *La France*, the turbines being arranged in series similar to the Spanish battleships. A similar arrangement is being adopted in the Cunard liner *Aquitania* and in the two liners for the C.P.R. Company now under construction on the Clyde. The two large vessels building in Hamburg for the Hamburg-American Line will also have four-shaft arrangements of turbine machinery.

The combination arrangement of turbines and reciprocating engines has also given good results, a saving in coal consumption of about 12 per cent. being claimed compared with similar ships having quadruple-expansion reciprocating engines only. In 1908 the New Zealand s.s. *Otaki* was installed with a three-shaft arrangement, in which the wing shafts were driven by triple-expansion reciprocating engines arranged to exhaust into a low-pressure turbine driving the centre shaft, and a similar arrangement of machinery has been adopted in the White Star liners *Laurentic* and *Olympic*, and in several other vessels. The slow-speed French Transatlantic liner *Rochambeau* has a combination arrangement of machinery driving four shafts, in which two sets of reciprocating engines are installed on the

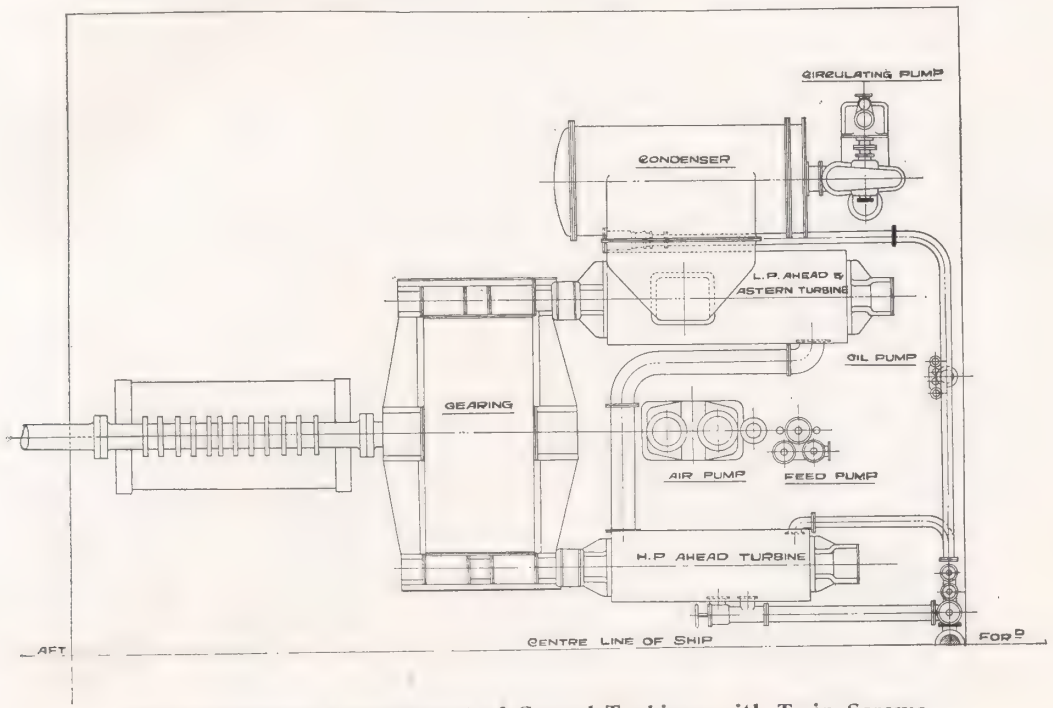


Fig. 2.—General Arrangement of Geared Turbines with Twin-Screws.

inboard shafts and the exhaust steam from each is passed into low-pressure turbines arranged on the outboard shafts. Two ships with similar installations are building for Spanish owners.

In 1910 the s.s. *Vespasian* was fitted with a geared installation of turbines. This ship has been in regular commission for eighteen months, during which she has steamed about 32,000 knots. No trouble has been experienced with the gearing, and there is no appreciable wear on the teeth of either pinion, the original pair being still in use. Geared turbine machinery driving two shafts is being fitted in two fast destroyers now under construction, and in two channel steamers building for the London and South Western Railway Co. In the latter vessels each shaft will be driven by two fast-running turbines arranged in series and connected to a double helical gear wheel through pinions, as shown in Fig. 2.

NEW EXPERIMENTS UPON THE PHENOMENA OF FLOW AND THEIR APPLICATION TO STEAM TURBINES, CONDENSATION, AND REFRIGERATION.

By Professor E. Josse.

(*Schiffbautechnische Gesellschaft; November*, 1911.)

IMPORTANT experiments in connection with this subject have been recently carried out under the author's direction at the Engineering Laboratory of the Technical High School at Charlottenburg. The research work, in so far as it dealt with steam turbines, was conducted by Dr. Ing. Christlein, and had for its object the determination of the phenomena and losses associated with the flow of steam through pipes and nozzles and the circumstances under which the losses are reduced to a minimum. The losses were determined by measuring the so-called

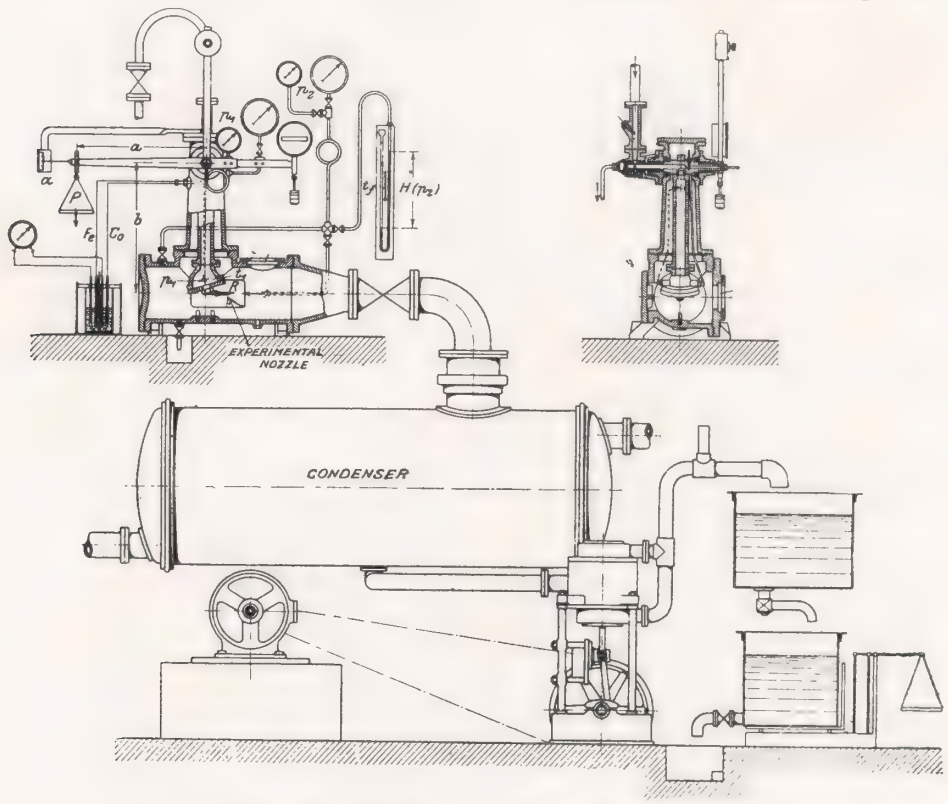


Fig. 1.—Arrangement of Apparatus used for Experiments.

reaction pressure, a method which had already been applied by other investigators (Rosenhain, Frederic, Kemble). The apparatus used for this purpose is shown in Fig. 1.

the actual speed of discharge w can be determined from (1), *i.e.*,

$$w = \frac{9{\cdot}81\,R}{G} \qquad (2)$$

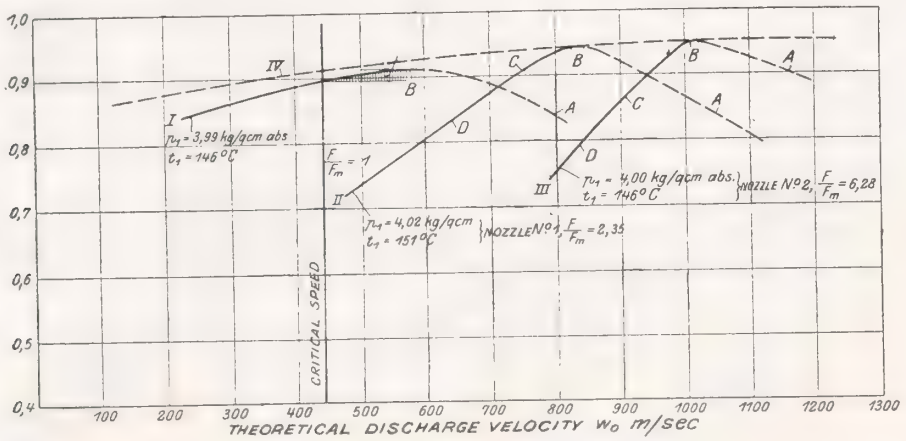


Fig. 2.—Curves of $\varphi = \frac{w}{w_0}$

The reaction pressure R, in kg., at the point of discharge is given by

$$R = \frac{G}{g}\,w \qquad (1)$$

where G = weight of steam flow in kg. per sec.
g = 9·81 metres per sec.
w = mean actual speed in metres per sec. in the direction of the axis of the nozzle across the section at which equality exists between the pressure in the jet and the pressure in the space into which the discharge takes place.

If R and G are determined by experiment, then

In the experimental apparatus the nozzle to be experimented upon is attached to a weigh beam and the reaction R is measured by weights P placed in the scale pan as shown in the illustration. Steam at the known initial pressure of p and temperature t is led to the nozzle, and discharges into the chamber A, in which there is a known pressure p_2. The theoretical discharge velocity w_0 is given by the formula

$$w_0 = 91{\cdot}53\,\lambda \qquad (3)$$

where λ is the heat drop in metrical units, *i.e.*,

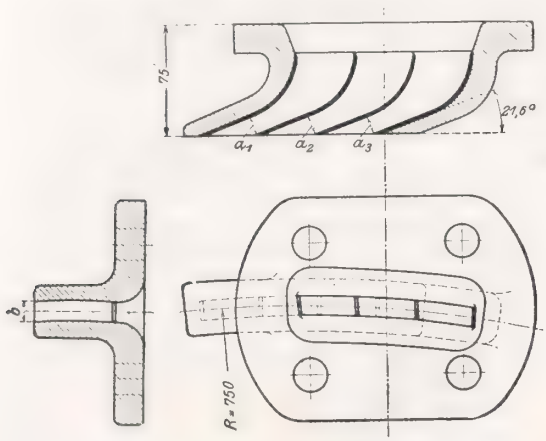


Fig. 3.—Nozzle with $\frac{F}{F_m} = 1$.

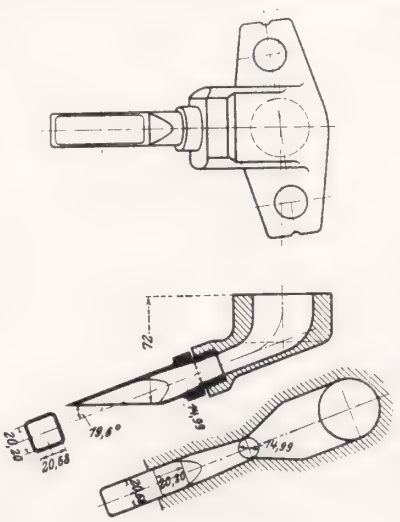


Fig. 4.—Nozzle with $\frac{F}{F_m} = 2{\cdot}35$.

the difference between the total heat of the steam at the entrance to the nozzle and the heat after frictionless adiabatic expansion to the back pressure p_2. w_0 can readily be obtained from the well-known Mollier diagram.

A correct knowledge of the ratio $\frac{w}{w_0}$, the so-called velocity coefficient φ, is of great importance in the design of turbines. Values of this coefficient as obtained for three different forms of nozzles, plotted in relation to the theoretical discharge velocity w, are given in Fig. 2.

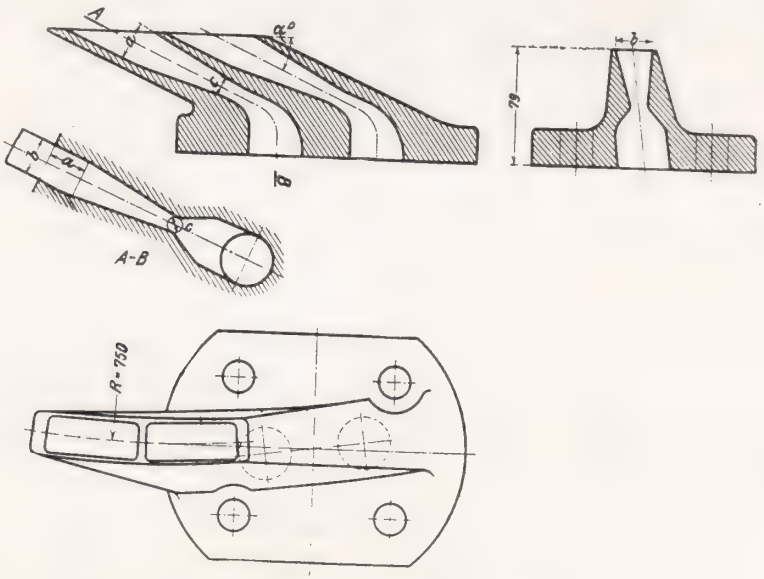


Fig. 5.—Nozzle with $\frac{F}{F_m} = 6{\cdot}28$.

Curve I. refers to a nozzle with parallel sides, and therefore having a ratio of $\frac{\text{outlet section}}{\text{narrowest section}}$, $\frac{F}{F_m} = 1$ (Fig. 3). Curve II. refers to a conically enlarged nozzle in which $\frac{F}{F_m} = 2{\cdot}35$, having the narrowest section circular and the outlet section rectangular (Fig. 4). Curve III. refers to a conically enlarged nozzle in which $\frac{F}{F_m} = 6{\cdot}28$, having the narrowest section circular and the outlet section rectangular (Fig. 5).

It will be noticed that each curve of $\varphi = f(w_0)$ has a definite maximum. It follows that the losses in the case of a particular nozzle will, under certain definite circumstances, be a minimum. Photographs taken of the steam flow through the observation holes in the chamber A showed that at the time of maximum efficiency the steam left the nozzle in a continuous jet with parallel stream lines, as shown by Fig. 6. At higher velocities vibration phenomenæ were observed, and also a deflection of the jet away from the axis of the nozzle. For such velocities Fig. 7 shows the phenomena observed in the case of a nozzle having the outlet section normal to the axis, and Fig. 8 the deflection of the stream lines in the case of a nozzle having an outlet section inclined to the geometrical axis.

An important conclusion to be derived from the experiments, as is shown by the enclosing Curve IV. in Fig. 2, is that the losses, in a nozzle properly proportioned for the pressure relation in question $\frac{p_2}{p_1}$, continuously decrease with increased speed of flow, which is just the opposite to existing opinion. A further surprising result is that the most favourable velocity in the case of an ordinary nozzle with parallel sides considerably exceeds the so-called critical velocity, while up to now it has always been assumed that to exceed the critical velocity with an ordinary nozzle would entail very considerable losses. The experiments show that the vibrations set up after passing the critical velocity have no great importance until much higher speeds are reached. The deflection of the jet in an oblique-ended nozzle has a much more harmful effect.

From Fig. 2 it will be seen that the greater the contraction of the nozzle in relation to the outlet area, the more sensitive it becomes to

differences in the velocity of the steam. This is a fact which should be borne in mind in connection with the regulation of the speed of turbines.

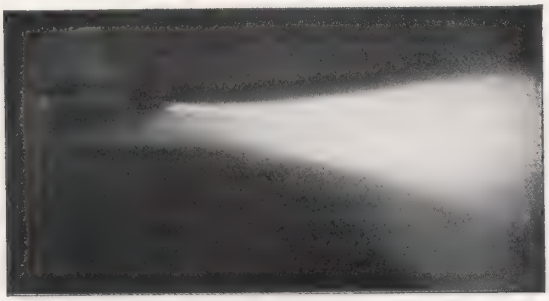


Fig. 6.—Nozzle $\frac{F}{F_m} = 2{\cdot}63$.

$p_1 = 5{\cdot}0$ *kg/cm² abs.* } Saturated
$p_2 = {\cdot}47$ *kg/cm² abs.* } Steam.

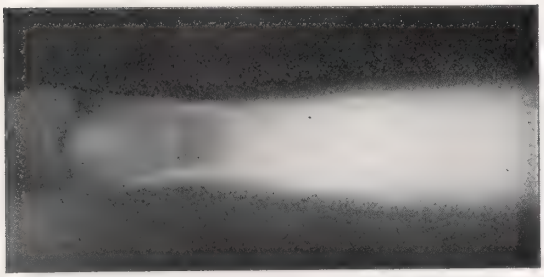


Fig. 7.—Nozzle $\frac{F}{F_m} = 1$.

$p_1 = 6{\cdot}03$ *kg/cm² abs.* } Saturated
$p_2 = {\cdot}482$ *kg/cm² abs.* } Steam.

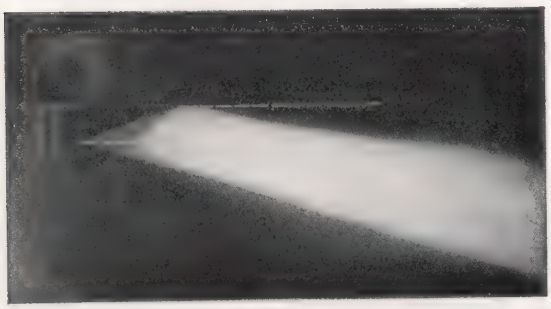


Fig. 8.—Nozzle $\frac{F}{F_m} = 2{\cdot}63$.

$p_1 = 9{\cdot}9$ *kg/cm² abs.* } Saturated
$p_2 = {\cdot}323$ *kg/cm² abs.* } Steam.

Experiments have also been made at the Charlottenburg Engineering Laboratory with regard to the reduction in velocity of the steam flow which takes place while the steam is passing through the turbine moving blades, the initial velocity w_1 being reduced to a value w_2 owing to the frictional and other resistances experienced. The total loss is expressed by the speed coefficient $\psi = \frac{w_2}{w_1}$. The values of ψ obtained in the above experiments (see Fig. 9) show that its maximum value, and therefore minimum loss, occurs at the so-called critical or "sound" velocity of the flowing medium.

With regard to the effect of the experiments upon the actual design and construction of turbines, the author considered that the improved thermal efficiency which the experiments showed to be obtainable would lead to simplification in design and lower first cost for an engine of given

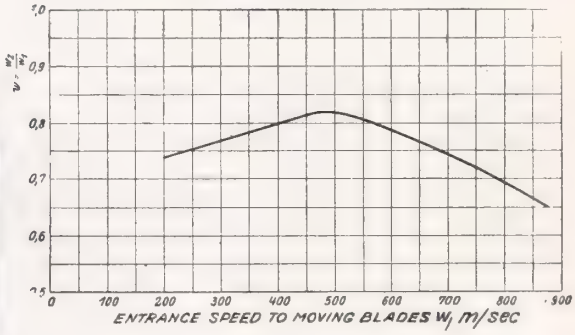


Fig. 9.—Curves of $\psi = \frac{w_2}{w_1}$.

power. The number of stages required would be reduced and the engine thereby shortened, while at the same time the velocity of rotation would be increased.

The efficiencies which have been obtained with present-day large turbines on land are shown by the full line in Fig. 10. The efficiencies which Professor Josse considers it will be possible to attain by improvements on the lines suggested by the experiments are shown by the upper dotted curve, and leave little to be desired. With the slow-running turbines required to suit the propeller in marine work, such high efficiencies are, however, out of the question. The improvements which the author has been able to effect in quick-running turbines, therefore, greatly strengthen the case for the adoption of such turbines in conjunction with a speed-reduction gear for marine work. Such a reduction gear is available in the Föttinger hydraulic transformer, which in the latest designs has an efficiency of 88 to 91 per cent. To show the superiority of quick-running turbines in conjunction with Föttinger hydraulic transformers, the author gives the following comparison :—

1.—Direct propulsion by slow-running turbines :—

Efficiency of slow-running multiple-stage turbine $\eta_g = \cdot 60$.

Efficiency of quick-running propeller $\eta_p = \cdot 65$.

Total efficiency of the installation $\eta = \cdot 39$.

In the case of the steam turbine the energy of the steam jet is converted into mechanical work. Another distinct application of jet or injection action is to the sucking-up and compression of gases. In conjunction with Dr. Ing. Gensecke, the author has applied this principle with success to the design of condensers in order to remove the

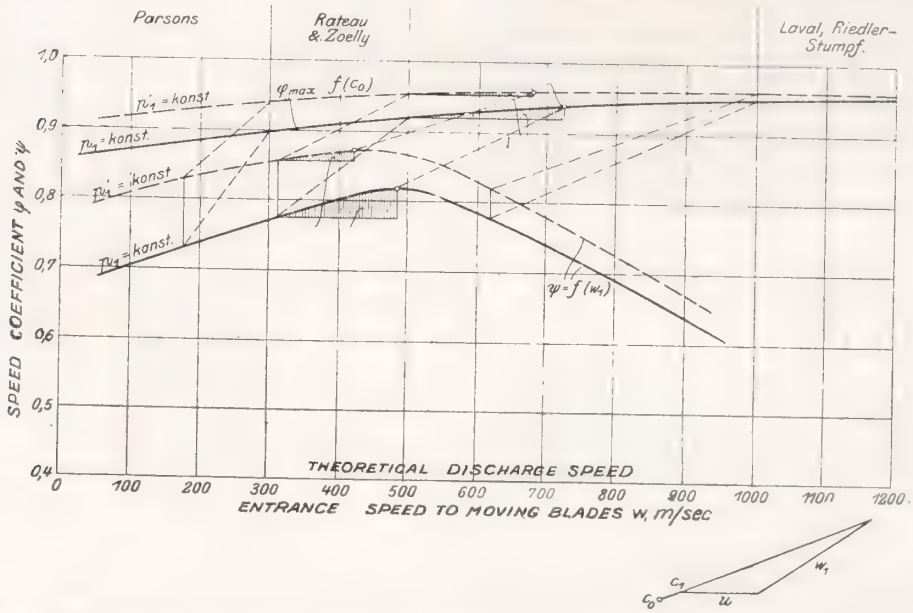


Fig. 10.—Comparison of present-day Values of φ and ψ with possible Values suggested by the Experiments.

2.—Propulsion by Föttinger transformers and quick-running turbines :—

Efficiency of quick-running turbine with suggested improvements $\eta_g = \cdot 75$ to $\cdot 78$.

Efficiency of transformer $\eta_t = \cdot 85$ to $\cdot 90$.

Efficiency of slow-running propeller $\eta_p = \cdot 78$.

Total efficiency of installation $\eta = \cdot 50$ to $\cdot 54$.

This comparison shows a superiority for the speed-reduction gear of about 30 per cent.

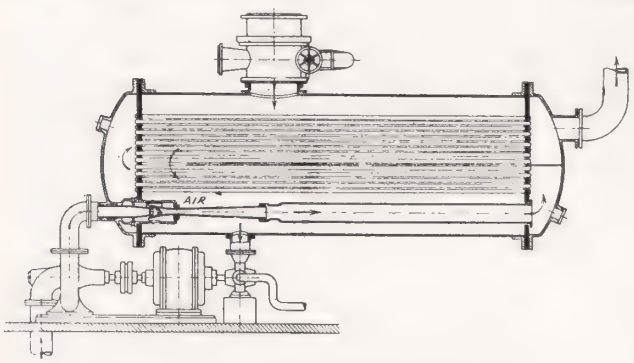


Fig. 11.—Josse-Gensecke Condenser for 250-K.W. Parsons Turbine.

gaseous products of condensation. The inventors utilize for this purpose the circulating water, or in special cases a steam jet in conjunction with the circulating water, or a steam jet alone. A surface condenser on their system is illustrated in Fig. 11, from which the simplicity of the design will be apparent. The jet is gradually increased in velocity in a nozzle, and then passes into a diffuser, sucking in with it the air and steam mixture which is to be removed from the condenser, as indicated by the arrows in the illustration. In the diffuser the kinetic energy of the jet is partly re-transformed into pressure energy in order that the medium may overcome the required pressure.

The application of jet action to condensers has long been suggested, but the author and his colleague have been the first to devise an efficient form of nozzle and diffuser, as the result of long experiment. The author then describes a number of installations which have been carried out on this system, and

from which very satisfactory results have been obtained.

The Josse-Gensecke jet refrigerating engine was developed from the experience gained by the inventors with condenser installations. It was found that steam flowing at a high velocity is a most satisfactory agent for taking up large volumes of steam or gas from the highest vacuum and for carrying away and compressing the same. The discovery of this fact enabled the inventors to successfully solve a further problem, *viz.*, the production of cold by the vaporization of water. This was accomplished by exhausting water vapour from a water tank under very low pressure (about ·006 kg./sq. cm.) by means of a steam jet of very high velocity. The water in the tank was thereby brought to give off vapour at a low temperature, and in doing so to withdraw heat from its surroundings. A refrigerating engine on this system is illustrated in Fig. 12. It will be seen that the apparatus is very simple and has no moving parts. The degree of cold produced is determined by the limit to which the pressure in the vaporizer can be reduced. This limit is about ·005 atm. abs., corresponding to a temperature of 23° F. The author has lately discovered,

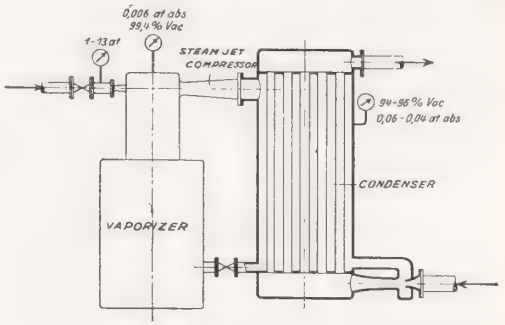


Fig. 12.—Josse-Gensecke Jet Refrigerating Engine.

however, that by the addition of a certain small quantity of air along with the jet, and with the same pressure in the vaporizer, temperatures down to about 2° F. can be obtained.

ELECTRIC DRIVES FOR SCREW PROPELLERS.

By Mr. H. A. Mavor.

(*British Association, Section G; September*, 1911.)

THE advantages claimed by the author for the electric transmission of power between the power generator and the propeller of a ship are as follow :—

1.—Prime movers which attain great economy at a high rate of revolution, such as the steam turbine and, to a smaller extent, the internal-combustion engine, can be adopted in conjunction with propellers running at the comparatively low rate of revolution at which the present-day propeller is most efficient, the speed reduction being accomplished by the electrical gearing.

2.—By its use a ready means is provided of reversing the direction of rotation of the propeller without changing the direction of rotation of the power generator.

3.—It provides means for changing the speed ratio between generator and propeller so as to permit of the power of the generator being developed under the most favourable conditions at all speeds of the ship.

4.—It provides means for applying the power of one or more engines to one or more propellers, so that the power-generating units may be of different types and so disposed as to give the highest efficiency; and when they are not required, as will happen when the ship is running at a reduced speed, they may be stopped.

The property of combining the power of more than one engine for application to one or more propellers is the special feature of the author's inventions as distinguished from the ordinary methods of electrical engineering. Alternating three-phase currents are used, and interlocking devices connect mechanically the main reversing switches with the existing switches, so that no change can take place in the connections while they are passing currents. The system has been thoroughly tested in the small experimental vessel *Electric Arc*, and is being installed in the steamship *Frieda*, now under construction for American owners. This vessel, of which a general arrangement is given in Fig. 1, has been specially designed for the transport of bulk freights between the Gulf of Mexico and New York City. She is to be 300ft. long, and will carry about 5,000 tons deadweight at a mean loaded speed of 12 knots at sea. The propelling machinery is placed aft, arranged as in Fig. 2, and consists of a turbo-electric outfit for 1,500 K.W. three-phase 50 cycles alternating current when running at 3,000 revs. per minute. The

current is led to a three-phase motor, which is keyed direct to the main propeller shaft, and is capable of developing 1,900 brake horse-power at a speed of about 84 revs. per minute. Steam at 200 lb. per square inch is supplied by two Scotch boilers fitted with Howden's forced draught and arranged to burn liquid fuel. The installation is said to weigh and cost less than a normal equipment, and a saving in fuel consumption equal to 10 tons of coal per day is expected. Proposed installations embodying electrical transmission in the case of a 245-ft. canal barge and a 525-ft. collier for the United States Navy are also described in the paper.

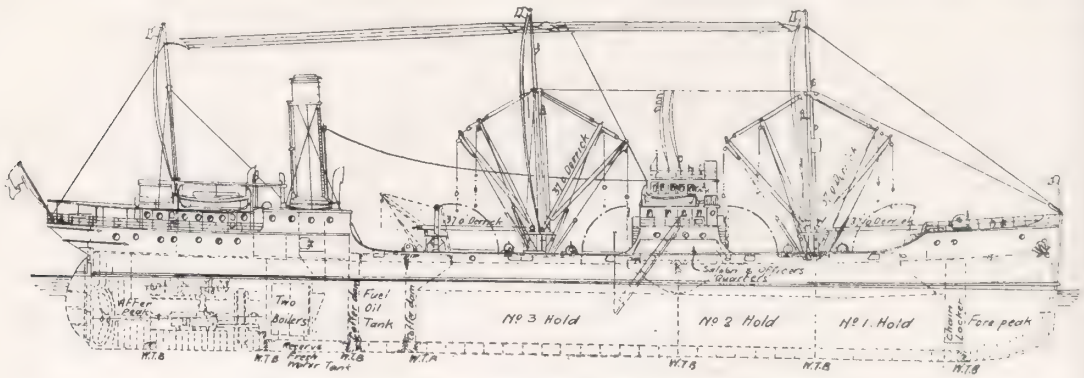


Fig. 1.—General Arrangement of the S.S. "Frieda."

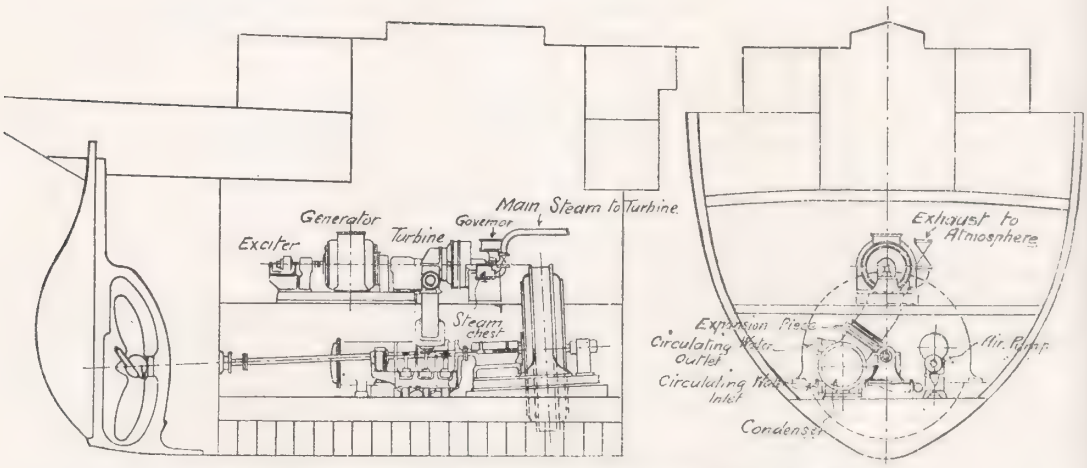
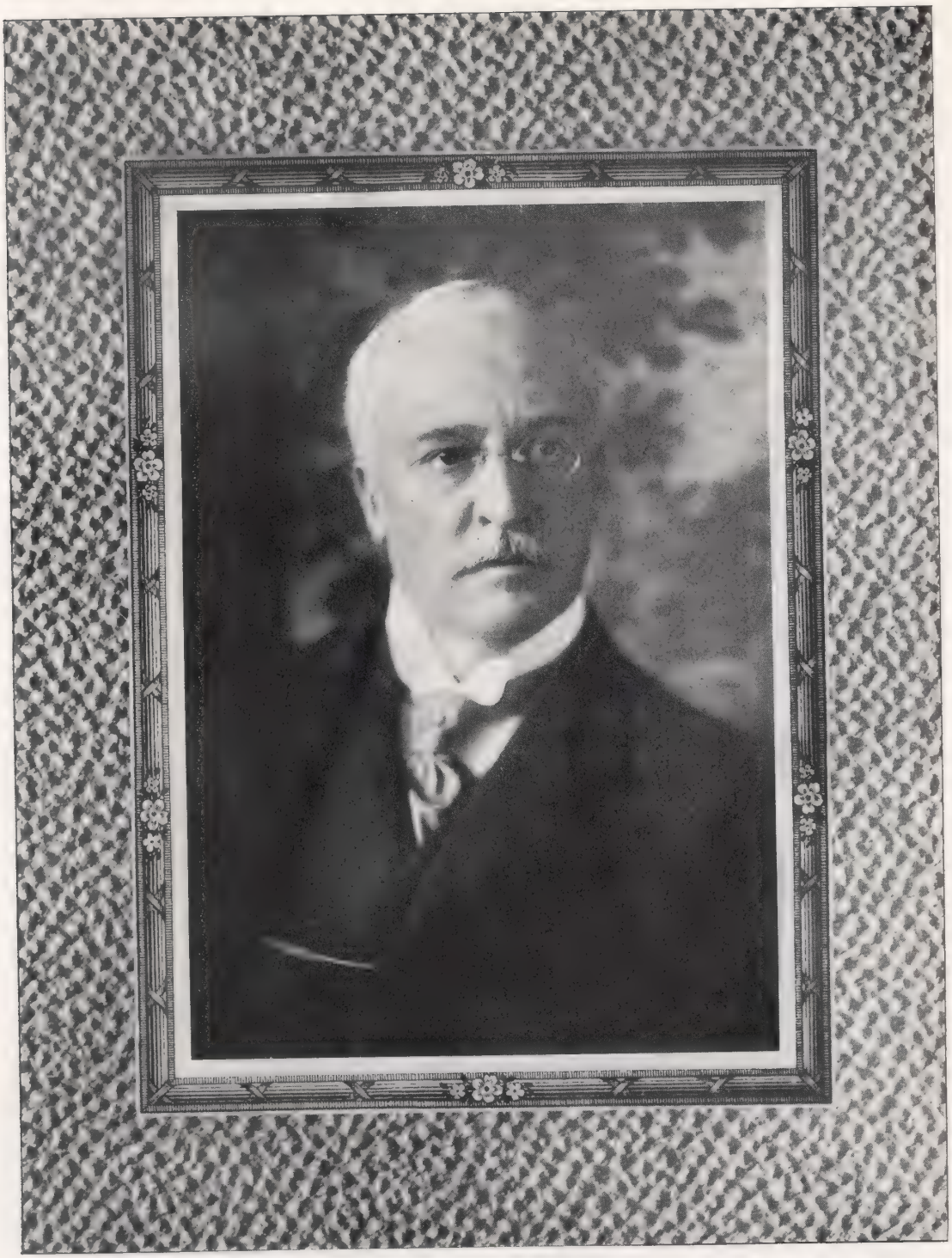


Fig. 2.—Arrangement of Propelling Machinery, S.S. "Frieda."



HERR DR. RUDOLF DIESEL.

Dr. Rudolf Diesel.

Internal-combustion Engines.

THE DIESEL OIL ENGINE AND ITS INDUSTRIAL IMPORTANCE, PARTICULARLY FOR GREAT BRITAIN.

By Dr. Rudolf Diesel.

(*Institution of Mechanical Engineers; March 15th*, 1912.)

THE first experimental Diesel engine was constructed by the author in 1893. The first reliable Diesel engine, a vertical engine of 18 H.P., was finally put into working order at the Augsburg Works in 1897 after four years of difficult and laborious experimental work, and, in the opinion of experts, gave better heat utilization than any known kind of heat engine. From the experience gained subsequently by working many engines, by gradual improvements in the construction and manufacture, and by

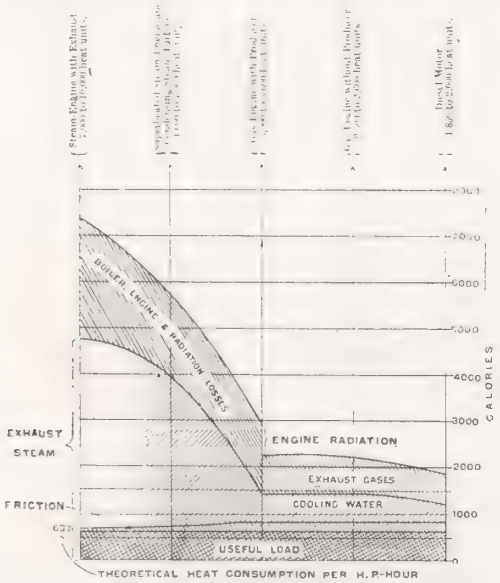


Fig. 1.—Heat Consumption of different Heat Engines per B.H.P.-hour.

increasing the sizes, the thermal or indicated efficiency of the Diesel engine to-day reaches 48 per cent., and the effective or brake efficiency reaches in some cases 35 per cent. of the heat value of the fuel. A comparison of the heat utilization for 1 B.H.P.-hour in different kinds of prime movers is shown in Fig. 1.

As regards the fuel supply for Diesel engines, the author states that it has been proved by recent geological researches that there is probably in the globe as much, or perhaps even more, mineral liquid fuel than coal, and that the oil-fields are more conveniently distributed than the coalfields as regards geographical positions. Moreover, the by-products of coal distillation and coke plants, such as tar and creosote oils, can be used in the Diesel engine with equally satisfactory results, provided certain precautions are taken in their preparation. It is also possible to burn fat vegetable oils and animal oils in the Diesel engine without any difficulty, thus making it certain that motive power can still be produced from the heat of the sun, even when all mineral stores of solid and liquid fuel are exhausted. In some tests made by the author for the French Government with Arachis or earthnut oil, the consumption obtained was ·53 lb. per B.H.P.-hour. As this oil can also be used for lubrication, the Diesel engine becomes a really independent engine for the tropics. Table I. contains a list of suitable oils and their qualities prepared at the Swiss Fuel-testing Laboratory of the University of Zurich.

Table I.

1.—*Normal oils which can always be used :—*

(*a*) Mineral oils freed from benzine (gas oils)	Hydrogen over 10 per cent. Calorific power over 10,000 cal. (39,680 B.Th.U.). No solid impurities.
(*b*) Lignite tar oils	Hydrogen over 10 per cent. Calorific power over 9,700 cal. (38,489 B.Th.U.)
(*c*) Fat oils from vegetable or animal sources, such as earthnut oil, castor oil, fish oils, etc.*...	Scarcely any researches have been made on these. Earthnut oil has 11·8 per cent. hydrogen, and calorific power 8,600 cal. (34,124 B.Th.U.).

2.—*Oils which can be used only with the aid of special apparatus :—*

(*a*) Pit coal-tar oil.

(*b*) Vertical-oven, water-gas and oil-gas tars, probably also coke-oven tars, the tests on which have not yet been completed.

General characteristics:

Hydrogen not over 3 per cent.
Amount of free carbon not over 3 per cent.
Residue on coking not over 3 per cent.
Calorific power not under 8,600 cal. (34,124 B.Th.U.).

*This class of oil has been added by the author from his own investigations of earthnut oil.

3.—*Oils which cannot be used :—*
Tars from horizontal or inclined retorts.

Dr. Diesel proceeds to give a brief historical account of the development of the Diesel engine from 1897 to the present day. As this involves constant reference to the four-stroke and two-stroke cycle engines, he first deals with the principal movements in these two systems, which are illustrated in Figs. 2 and 3, the corresponding indicator diagrams being given in Figs. 4 and 5. With regard to the relative advantages of the two systems, the author states that the four-stroke cycle engine still has a better combustion and a more economical fuel consumption, and is, above all, simpler in its method of working. It thus remains the standard perfect engine, and

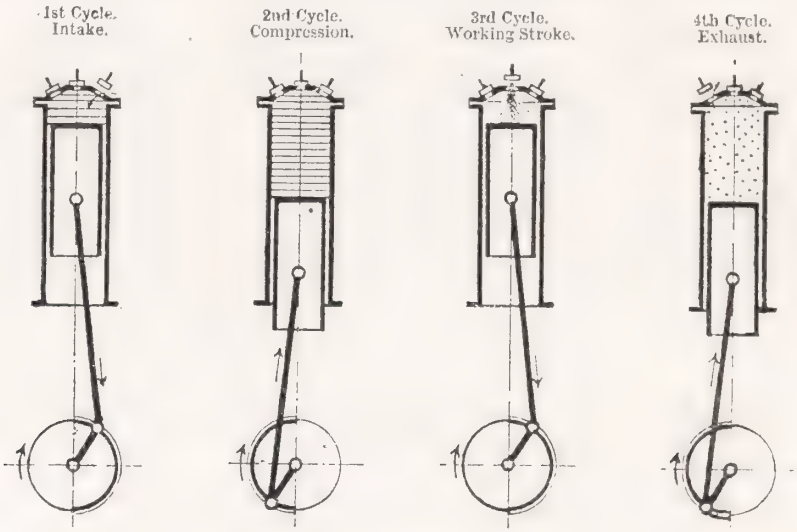


Fig. 2.—Four-stroke Cycle.

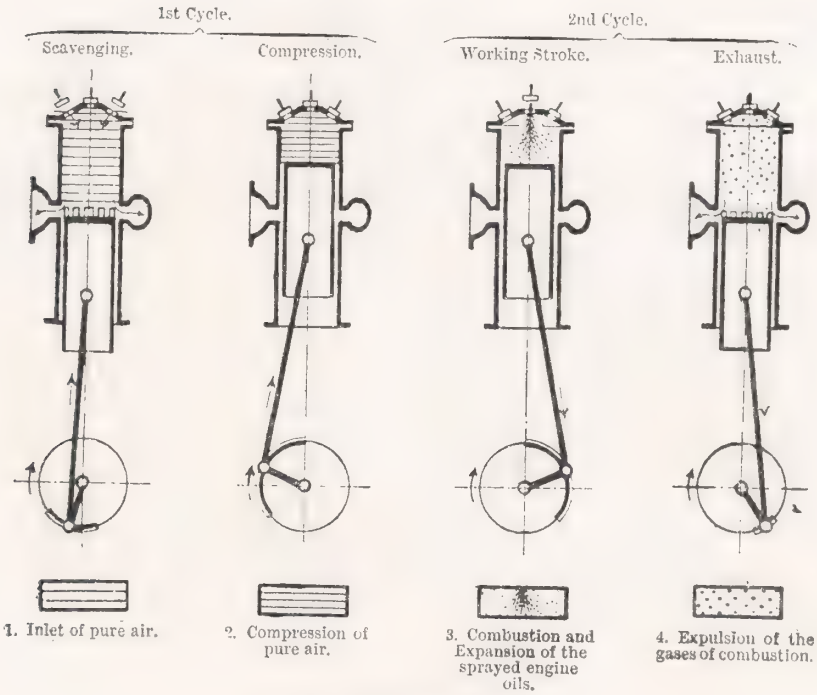


Fig. 3.—Two-stroke Cycle.

still predominates for medium-sized stationary plants up to 500 or 600 H.P. On the other hand, the two-stroke cycle engine, with its smaller cylinders, has now come into favour for stationary plants of higher horse-power, and, as a marine engine, has become the only standard type.

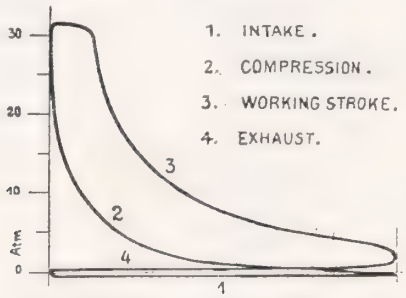


Fig. 4.—Four-stroke Cycle.

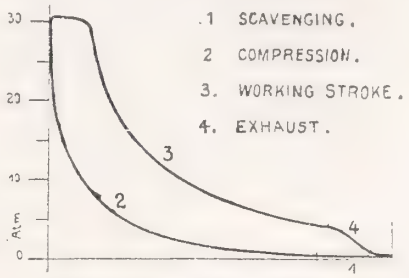


Fig. 5.—Two-stroke Cycle.

Although it may never equal the four-stroke cycle engine as regards thermal efficiency, its initial cost is so much lower that its slightly higher fuel consumption will be more than counterbalanced by the greater interest and amortization on the higher priced four-cycle engine. A typical four-stroke cycle high-speed engine, made by Messrs. Sulzer Brothers in 1909, is shown in Fig. 6.

annular piston, is placed under each combustion cylinder. Both engines are single-acting. Their relative merits can only be settled by experience. Another type of two-cycle engine has recently been brought out by Professor Junkers on the lines of the old Öchelhauser gas engine, with two pistons working in opposite directions in one cylinder but acting on the Diesel principle.

The first marine Diesel engine of 20 H.P. was constructed in 1902-3 in France, for use on a canal boat, by the French engineers Adrien Bochet and Frédéric Dyckhoff, in conjunction with the author. After that date the evolution of the Diesel marine engine steadily continued, chiefly on the demand

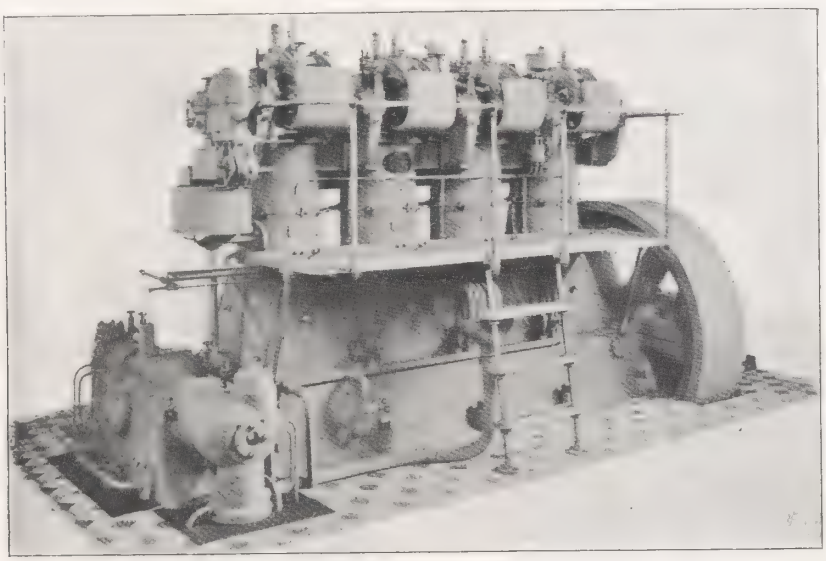


Fig. 6.—Four-stroke Cycle High-speed Engine.

Two considerably different types of two-stroke cycle engines have so far been competing. The first type is that made by Messrs. Sulzer Brothers with separate scavenging pump (Fig. 7). In the second type, that made by the M.A.N. (Fig. 3, page 138) the scavenging pump, which has an

of the French submarines and Russian river boats. These engines were on the four-stroke cycle system and not originally reversible. In the most favourable case (*Delproposto*) the propulsion of the vessel was performed directly by the engine, while the manœuvring and slow driving were done

by means of electricity. The first reversing marine two-stroke cycle engine was built in 1905 by Messrs. Sulzer Brothers at Winterthur and fitted to a vessel for service on Lake Geneva.

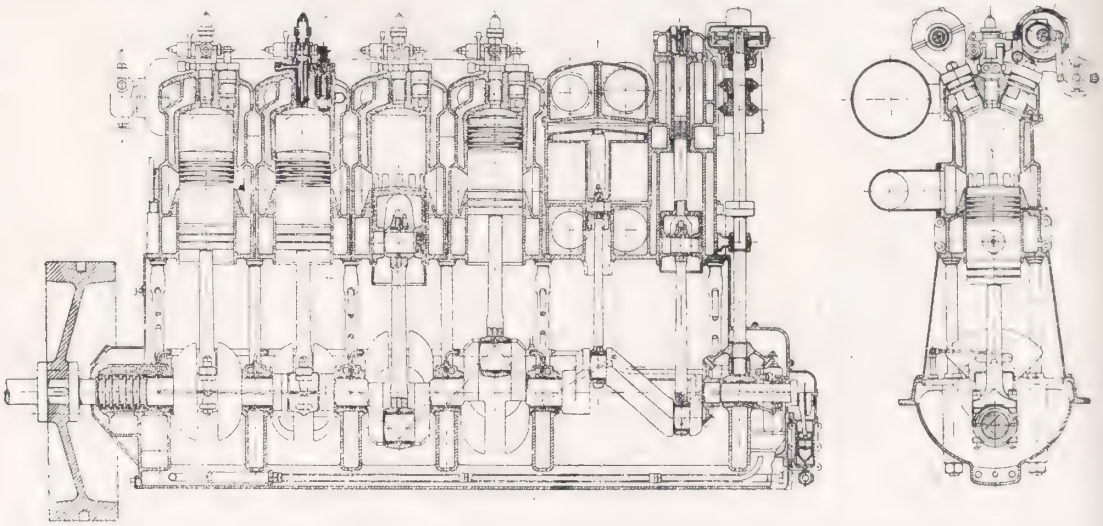


Fig. 7.—Single-acting Two-stroke Cycle Engine, with separate Scavenging Pump.

The first four-stroke cycle reversible marine engine was built by Messrs. Nobel Brothers at St. Petersburg in 1908 and fitted to a Russian submarine. Great mechanical complications were

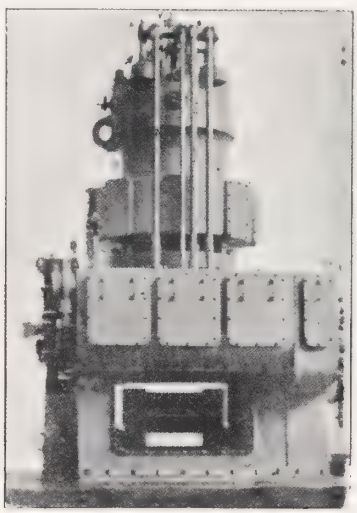


Fig. 8.—Single-cyl. Double-acting Two-stroke Cycle Engine, 1,200 H.P.

at first associated with the reversing gear of the four-stroke cycle engine, but in some recent examples of this type the reversing arrangements have been much simplified.

Quite recently double-acting vertical two-stroke cycle Diesel engines have been built. One such engine, made by Messrs. Carels Frères, having one cylinder of 1,200 H.P., is shown in Fig. 8. No information with regard to the performance of this engine has yet been published. At the Nürnberg Works of the M.A.N. important experiments with double-acting two-stroke cycle engines are also being carried out. In these works prolonged official tests of a three-cylinder double-acting two-cycle engine of 850 H.P. were made in August, 1911, and the results proved highly satisfactory. At the present time a double-acting two-cycle three-cylinder engine of 2,000 H.P. per cylinder is being tested at Nürnberg. The dimensions of these cylinders are 31½in. diameter and 41⅜in. stroke, with 160 revolutions per minute. This engine has already yielded considerably more than 2,500 H.P. per cylinder, so that an engine unit of this kind with six cylinders would give 15,000 H.P., or 45,000 H.P. for a vessel with three propellers. At Messrs. Krupp's Germania Works cylinder units of 2,000 H.P. double-acting two-cycle are being tested at present, also at Messrs. Sulzer's works a single-acting two-cycle cylinder of 2,000 H.P.

As to the special importance of the Diesel engine for Great Britain, the author maintains :—

1.—Enormous savings can be effected in coal, and the exhaustion of the coalfields delayed by first transforming the coal into coke and tar oils by distillation, and using the latter, after the valuable chemical by-products have been extracted, in a much more economical way in the Diesel engine than is the case by burning coal direct on the grate of a steam boiler.

2.—As Great Britain has the largest Colonial Empire in the world, it is to her of the greatest importance to utilize the natural mineral oils found in many of her colonies, while the fact that vegetable oils can be used for the Diesel engine is of great importance for tropical colonies where coal and mineral oils are scarce.

3.—With regard to marine engines, it is unquestionable that one of the greatest evolutions of modern industry will be connected with this development of the Diesel engine, and Great Britain, as the greatest shipping nation of the world, will derive the greatest advantage from it.

In an appendix the author gives a list of about 300 vessels propelled by Diesel engines, built or in course of construction up to November, 1911, and states that since that date numerous other orders have been placed. In another appendix some results obtained with Diesel-engined vessels are given, so far as such are available, and show that the fuel consumption is only from one-fifth to one-fourth the weight of coal which would be used on a steamship of equal power.

CRUDE-OIL MARINE ENGINES.

By Mr. James H. Rosenthal.

(*British Association; September*, 1911.)

THE two types of engines which are particularly dealt with in this paper are the Bolinders engine, as an example of a crude-oil reversible marine engine which is not on the Diesel principle, and the Diesel engine proper. The former type is one which is essentially suitable for small craft, and it has been largely fitted on fishing vessels and barges. The sizes generally used run from 10 to 30 H.P., although sizes up to 150 H.P. have been made and are running. Fig. 1 shows this engine, which is of the two-cycle type. To start the engine a blow lamp is inserted through the cover of the ignition chamber, or hot pot, *E*. When this chamber is heated to a dull red, the engine is turned, either by hand in the smaller sizes, or by compressed air in the larger sizes, and a charge of fuel is admitted through the fuel valve *F*. The fuel is ignited by contact with the heated chamber *E*, which forms the cylinder head and burns in air previously admitted by the uncovering of the passage *H* and compressed during the upward

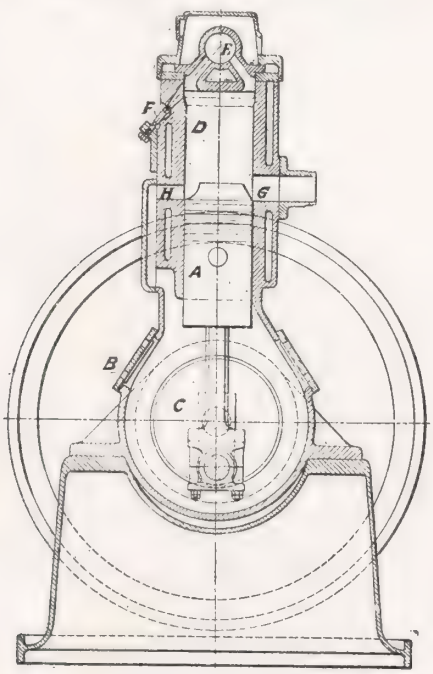


Fig. 1.—Bolinders Two-cycle Engine.

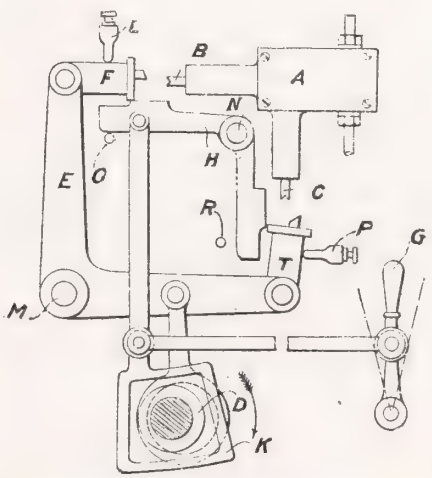


Fig. 2.—Reversing Gear, Bolinders Engine.

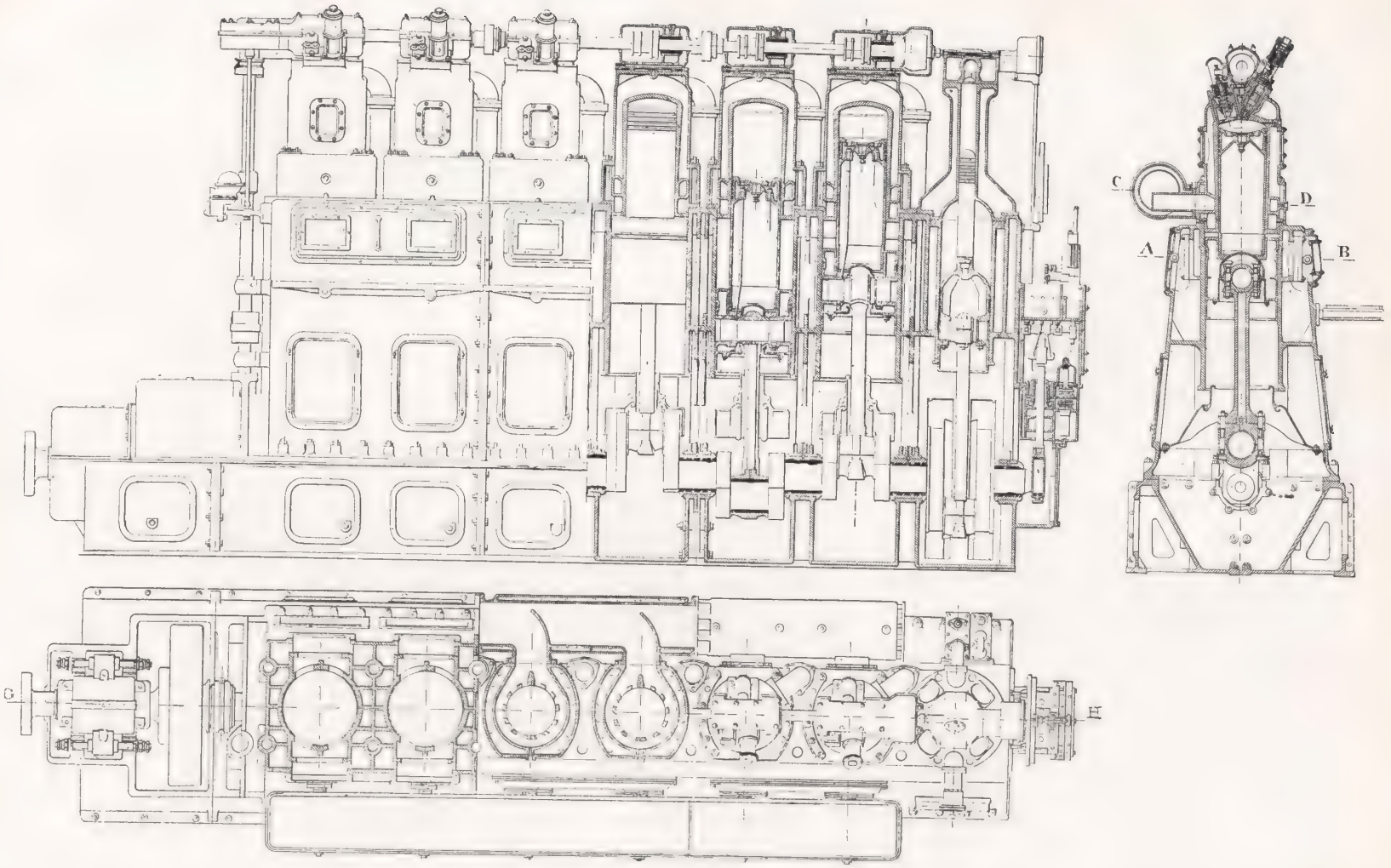


Fig. 3.—Nürnberg Two-cycle Single-acting Marine Engine.

stroke of the piston. As the piston approaches the end of the downward stroke, it uncovers the port *G*, through which the gases commence to exhaust, their expulsion being completed by the subsequent admission of scavenging air, which is slightly compressed in the crank case as the piston moves downwards.

The reversing operation is illustrated by Fig. 2. In this diagram *A* is the fuel pump case and *B* and *C* are the fuel pump plungers. These are operated by a toggle *F* when running normally, and by a toggle *T* when reversing, *F* and *T* being connected to the opposite ends of a bell crank *E*, which is operated by an eccentric on the engine crankshaft. To reverse the engine the link *K* is pulled on to the friction disc *D* by the reversing lever, thereby pushing the toggle *F* out of action and toggle *T* into action by means of the bell crank H. The fuel pump *B*, which in normal running injects the fuel at the top of the stroke, is thus temporarily thrown out of action and pump *C* substituted. The pump *C* is timed to inject a charge before the piston reaches the top of the stroke, thus checking the upward motion of the piston and driving it down in the reverse direction. When the engine has reversed, the bell crank *E* resumes its former position and fuel is again injected at the top of the stroke for normal running.

The type of engine suitable for larger powers is that on the Diesel principle, on what is called the "Diesel cycle of combustion." This cycle of combustion is based upon the fact that when air is compressed to 450 lb. per square inch it attains a temperature of about 1,000° F., and will then ignite atomized crude oil with which it is brought into contact. The engine put forward as an example of the latest development of the reversible two-cycle crude-oil marine engine working on the Diesel principle is that made by the Maschinenfabrik Augsburg-Nürnberg. It is constructed single-acting and double-acting.

The Nürnberg marine oil engine found its first introduction on board submarines. For use in such vessels it is run at comparatively high speeds, varying from 550 r.p.m. for a 200-H.P. engine to 420 r.p.m. for an engine of 900 H.P. For lightness the framing and bedplates are made of manganese bronze, and gear covers, etc., of aluminium, the average weight of fairly large sizes being 35 to 37 lb. per brake horse-power. The same principle of engine has been used, but built with less regard for saving in weight and at less cost per horse-power, for slower running commercial installations. These engines run at speeds from 360 to 160 r.p.m.

The reversible double-acting two-cycle engine

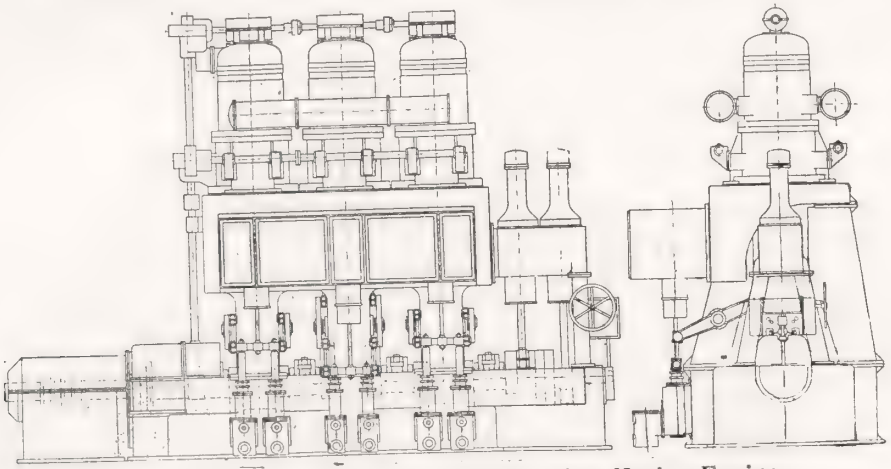


Fig. 4.—Nürnberg Two-cycle Double-acting Marine Engine.

is on the eve of being tried commercially for long distances on a vessel of the Woermann Line, and on a vessel of the Hamburg-Amerika Line, both vessels being built by Messrs. Blohm & Voss. In the case of one vessel there are two engines of 1,000 H.P. each and in the other two engines of 1,500 H.P. each. The following are the particulars of one of the 1,000-H.P. engines :—

Number of cylinders	3
Diameter	18·9in.
Stroke	25·6in.
Revolutions of the engines	125
Diameter of crankshaft	12in.
Weight of engine	75 tons

Limits of speed, 40 to 125 revs. normally.

During shop tests this engine is said to have run quite satisfactorily up to 220 r.p.m.

Single-acting engines are made with from four to eight cylinders, according to the power required. A 300-H.P. two-cycle six-cylinder single-

acting engine is illustrated by Fig. 3. In this example the air compressor, which is an essential feature of all Diesel engines, is driven by an extra crank attached to the main engine and placed at the forward end of the cylinders, as are also the oil and water circulating pumps and the reversing gear. The cylinders are jacketed and cooled with sea or fresh water. The piston is of two diameters, the upper part being the working piston above which the combustion takes place. The lower and larger diameter works in a liner and forms the guide as well as the compressor for the scavenge air, which is stored in the upper part of the crank case, and from thence is led to the scavenge valves. The head of each cylinder is provided with a fuel valve with its injection air attachment, a scavenge valve, and an air starting valve, all of which are operated by cams placed on a crankshaft running along the tops of the cylinders, there being two cams for the starting valve, one for going ahead and one for going astern. The speed of the engine is varied by regulating the quantity of the charge of fuel oil. The compressed air for injecting the fuel is led from an air vessel which stands alongside the engines, and which is charged with air at about 800 lb. by the air compressor. The reversing of the engine is accomplished by the operation of either a lever or a handwheel on bell crank tuppets, which operate air valves. The consumption of fuel oil varies from ·52 lb. in small engines to ·45 lb. in large engines.

The double-acting engine is built with three working cylinders and separate scavenge-air cylinders, as the working cylinder pistons cannot in this case be stopped to form scavenging pumps. The scavenge cylinders can be placed in line with the working cylinders and their pistons worked from a continuation of the main crankshaft, or they may be placed on the side of the engine and worked by rocking levers from the main crossheads, as in the case illustrated by Fig. 4. The cycle of operations is precisely the same as in the single-acting engine, except that it takes place at each end of the cylinder alternately, and for this reason there are cam shafts both top and bottom of the cylinders. In order to get a better distribution of fuel oil and more even burning, the valves at the bottom are in duplicate. The reversing and starting gear is similar to that on the single-acting engines, but actuates starting air valves at each end of the cylinders.

RESEARCH AND EXPERIMENTAL WORK IN CONNECTION WITH THE CONSTRUCTION OF THE JUNKERS LARGE OIL ENGINE.

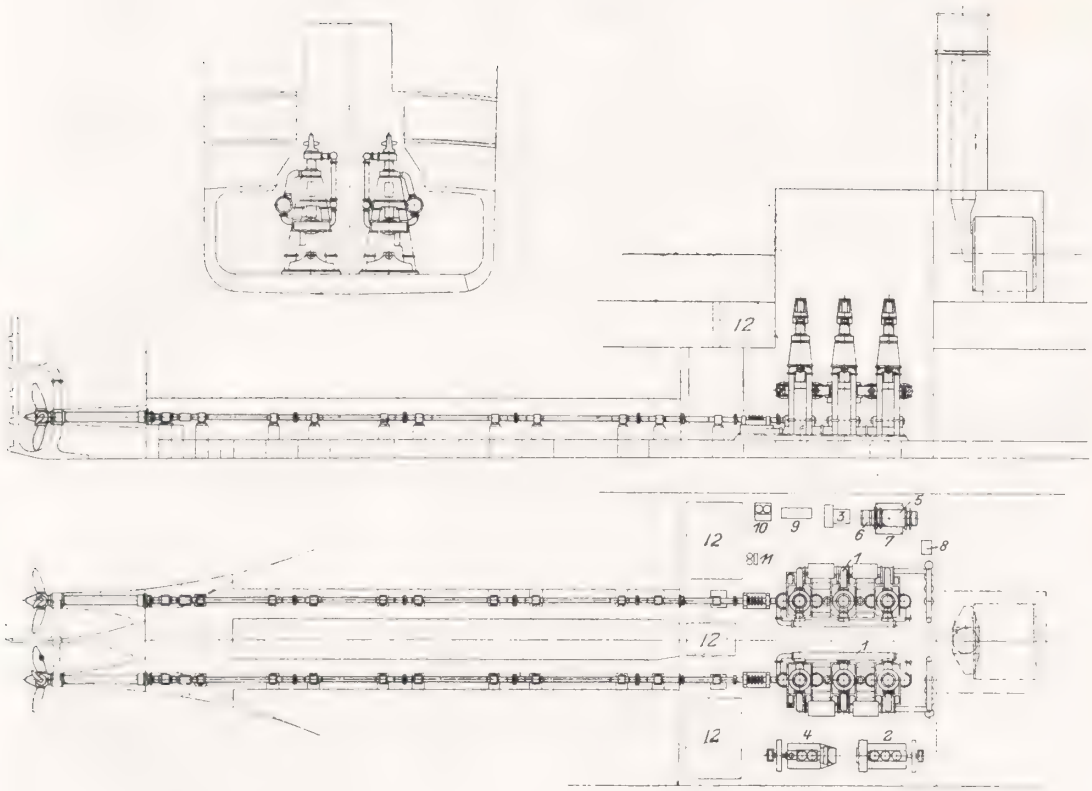
By Professor H. Junkers.

(*Schiffbautechnische Gesellschaft; November*, 1911.)

IN the course of this most valuable and interesting paper Professor Junkers first deals with his research work from the year 1889 onwards in connection with internal-combustion engines, and the gradual evolution of the type of large oil engine with which his name is associated. His first experiments, conducted in conjunction with Dr. von Ochelhauser, showed that with the two-cycle process it was possible, by the adoption of high compression, to secure a much greater fuel economy than that shown by the Otto motor. The greatest difficulties experienced were those due to the intense heat developed in the cylinders and the high stresses caused thereby in the cylinder walls. The author's experience showed that, for this reason, it is important to introduce the charge into the cylinders at as low a temperature and in as cleanly a state as possible, and to reduce openings and changes of section in the cylinder walls to a minimum. The transmission of the heat was found to depend largely on the gas pressure and eddy action, and the influence of these factors should not be neglected.

Protracted studies and experiments at the Gas-engine Laboratory of Dr. Ochelhauser and Professor Junkers in Dessau led finally to the construction of an experimental two-cycle double-piston gas engine, in which the foregoing principles were embodied and with which most successful results were obtained. The next step was to develop a type of oil engine based on the same principles, which would be suitable for ship propulsion. It was first necessary to prove that the experience gained with gas engines was applicable to oil engines working on the Diesel process, and in 1908 an experimental single-cylinder oil engine was built for this purpose. After long and tedious work, successful results were achieved. In 1910 this engine was altered to a tandem arrangement with two pistons working in each of two cylinders, and the research continued. In 1911 a still larger experimental engine, of 1,000 H.P., was installed at Professor Junkers' laboratory at Aix-la-Chapelle, with which experiments are still in progress.

Experience with the experimental engines has



1. Main Engines.
2. Reserve Compressor with Diesel engine.
3. Steam-driven Compressor.
4. Dynamo with Diesel engine.
5. Auxiliary Condenser.
6. Combined Air and Circulating pump.
7. Tank with float.
8. Steam feed pump.
9. Electrically-driven ballast pump.
10. Steam pump for boiler feed tank.
11. Steam pump for oil receiver.
12. Oil Fuel Tanks.

Fig. 1.—Arrangement of Junkers Engines on Cargo Vessel for Hamburg-Amerika Line.

been sufficiently promising to induce the Hamburg-Amerika Line to adopt Junkers engines for a vessel they are having built at the

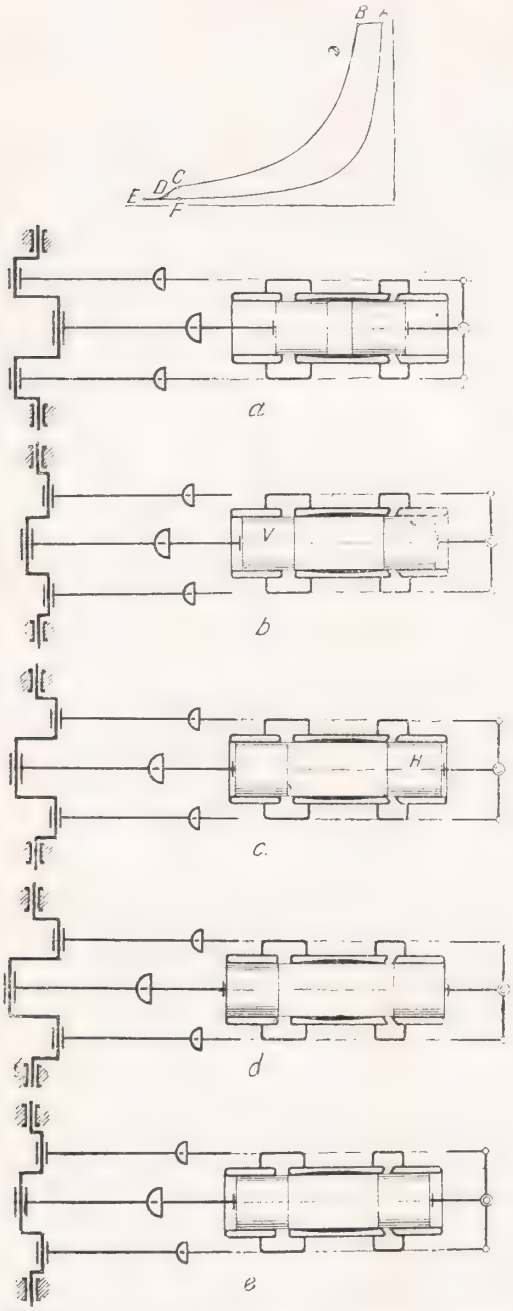


Fig. 2.—Cycle of Operations, Junkers Engine.

"Weser" Yard at Bremen. This vessel will have twin screws each driven by three vertical two-cycle tandem double-piston engines, the machinery being arranged as shown in Fig. 1. The total power of the installation is about 1,600 S.H.P. at 120 revolutions per minute. The cylinders have a diameter of 16in. and each piston a stroke of 16in. The design of the engine is shown in Fig. 3, and is similar to that of the experimental engines except that the tandem cylinders are placed vertically above one another instead of horizontally.

It will be seen from Fig. 3 that the engine is double-acting. The highest and lowest pistons actuate a central crank; while the other two pistons, by means of a return crosshead, work smaller cranks placed on each side of the middle crank and set at 180° from the latter. No covers are fitted to the cylinders, with the advantage that not only is the construction simplified but the cylinder walls are cooled during each return stroke by coming directly in contact with the atmosphere. The scavenging pumps and air compressor are worked from the return crosshead. Each cylinder has two fuel valves and one starting valve. The engine is governed and reversed by means of cams, which operate the fuel and compressed-air valves. The scavenging air is admitted and the exhaust gases are expelled through ports near the ends of the cylinders, which are opened and closed by the pistons themselves.

The action during one complete stroke will be understood from Fig. 2, in which, for the sake of clearness, one of the tandem cylinders is omitted. In Fig. 2*a* the pistons are shown in the innermost position at the end of a compression stroke, the small space between them being filled with highly compressed air at a high temperature, so that when the fuel is introduced in the form of a fine spray it ignites and burns for a short part of the outward stroke *A*-*B* under almost constant pressure. At *B* the fuel supply is cut off and the expansion of the products of combustion takes place from *B* to *C*. At *C* (Fig. 2*b*) the piston *V* exposes a circular port through which the gases exhaust. By the time the pistons have reached position *D* shown in Fig. 2*c* the pressure is approximately equal to that of the atmosphere. In this position the piston *H* lays open the circular port for the admission of the scavenging air at low pressure, which action continues until the pistons have passed the outer dead centre (Fig. 2*d*) and returned to the position shown in Fig. 2*e*, when the scavenging air port is again closed (*D E F* in diagram). At the point *F* the cylinder is filled with fresh air, which, not being in communication with the atmosphere, is compressed and thereby heated, until the point *A* is reached, when fuel is again injected and the process is repeated.

The last portion of the paper is devoted to the consideration of proposed applications of the

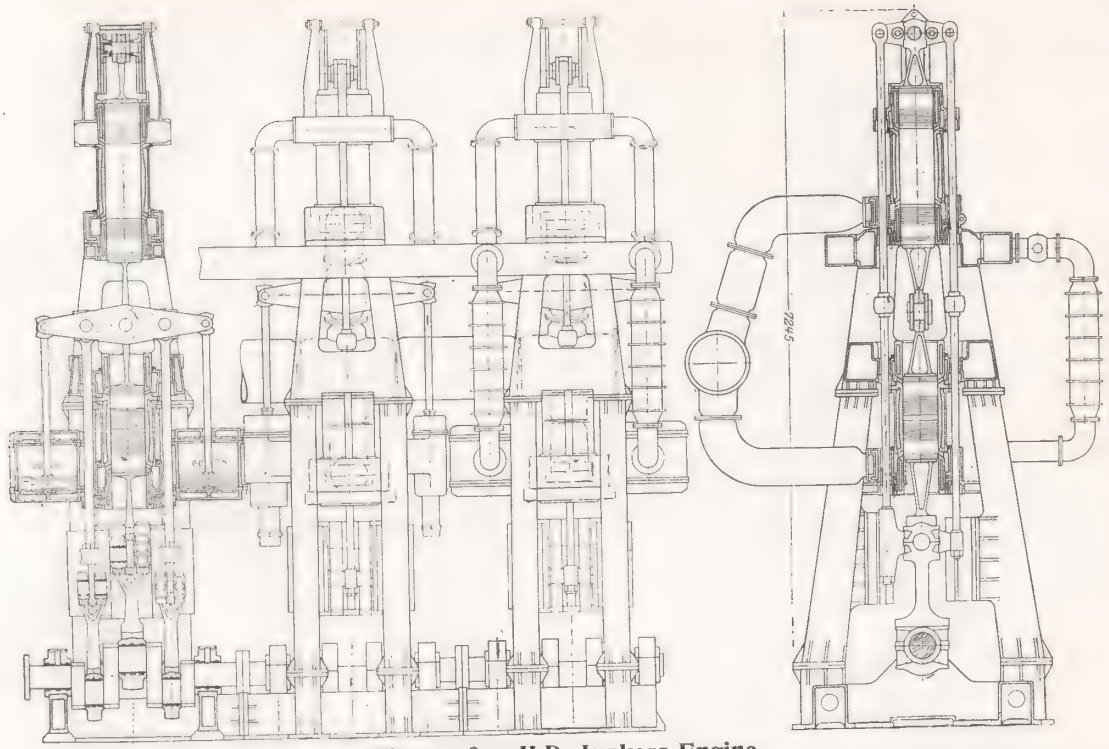


Fig. 3.—800-H.P. Junkers Engine.

Junkers engines to battleships, several outline arrangements being given for such vessels. The author thinks the horizontal type of engine will probably find most favour, owing to the desirability of keeping the machinery below the waterline.

THE GAS TURBINE.

By Herr H. Holzwarth.

(*Schiffbautechnische Gesellschaft; November*, 1911.)

AFTER briefly dealing with earlier inventions in this field, the author describes the Holzwarth-Junghans gas turbine, which is illustrated in Fig. 1. This turbine is an explosion turbine in which the combustion chamber is filled intermittently with gas and air mixture. Following each admission of this mixture occur ignition, explosion, and increase in pressure of the products of combustion, which expand through a nozzle into the actual turbine, followed by a thorough scavenging and cooling of the combustion chamber by means of fresh air. The cycle described is shown in the indicator diagram Fig. 2.

A series of ten combustion chambers *A* are arranged round the base of the engine illustrated and come into use regularly one after the other. The air and gas in the chambers *B* and *C* respectively are kept under low pressure by any convenient kind of compressor, and a vacuum is maintained at the exhaust port *J*. After the chamber *A* has been filled with air through the air inlet valve *D*, gas is blown in through the valve *E* and a thorough mixing of the air and gas takes place. During this period the nozzle valve *F* remains shut. Immediately after ignition the valve *F* is opened by the force of the explosion and the gases stream through the nozzle *G*, in which their speed is gradually accelerated till they impinge upon the turbine wheel *H*, by which their kinetic energy is absorbed, and then exhaust through the outlet *J*. After the passage of the gases the nozzle valve *F* is arranged to close slowly in order that the scavenging air admitted

at *D* may have plenty of time to cleanse the chamber *A* and cool the turbine rotor. The valve gear is driven by the shaft *M*, which is in turn driven from the main shaft by a helical wheel transmission.

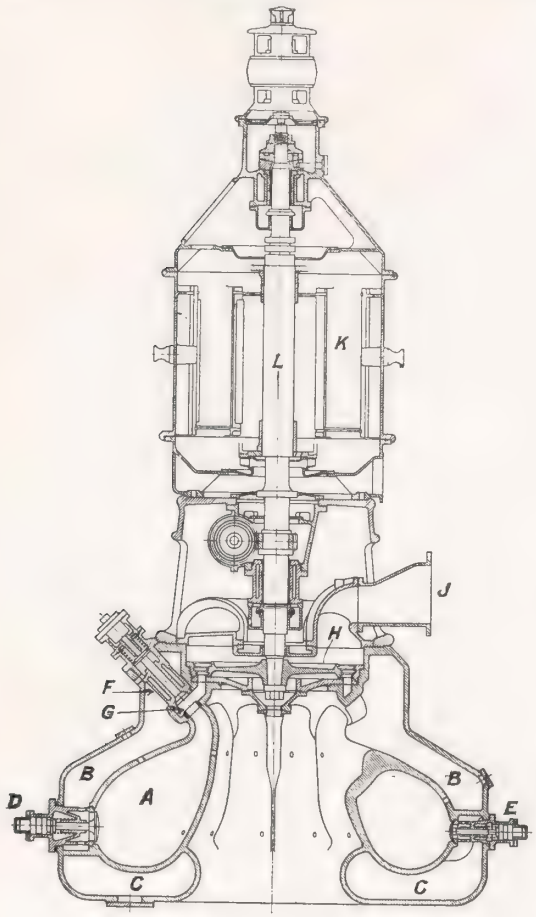


Fig. 1.—1,000-H.P. Gas Turbine.

The engine illustrated by Fig. 1 is the first working turbine constructed on this system. It is worked by suction producer gas of 124 to 135 B.T.U. per cubic foot, that is, a poor quality of gas, and has been built for a maximum output of 1,000 H.P. at 3,000 revs. per min. The turbine drives a dynamo *K*, which is placed above it on a vertical shaft *L*. With this engine exhaustive

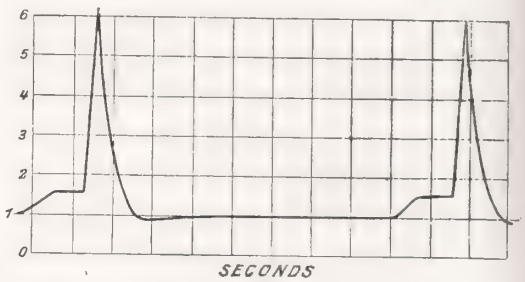


Fig. 2.—Normal Indicator Diagram.

experiments have been made and have resulted in the attainment of good efficiencies in the case of small charges and low pressures. For higher charges and pressures the results have not been so satisfactory, but the author feels confident of ultimate success after further experience has been gained. The inventors have found that the temperature of the air and gas mixture before ignition is most important and determines whether regular burning or detonation occurs.

The Holzwarth gas turbine can also be worked as an oil turbine using petroleum, gas oil, tar oil, etc., by the addition of fuel injection valves; worked, as in the Diesel engine, by compressed air.

THE OIL ENGINE IN THE GERMAN SEA-FISHING INDUSTRY.

By Professor F. Romberg.

(*Schiffbautechnische Gesellschaft; November*, 1911.)

THIS paper contains a very complete description of the various types of small oil engines which are being adopted on board German fishing boats. Full details and sketches of the various parts of the engines are given in each case. The first example chosen is a 24-H.P. two-cylinder "Brons" motor, made by the Gasmotorenfabrik Deutz, of Cologne, and illustrated by Fig. 1. The following are the main particulars of this engine :—

Cylinder diameter	8in.
Stroke	9½in.
Revolutions per minute, normal	340

The engine works on the four-stroke cycle, the action being similar to the Diesel process, with the exception that, owing to the simpler method of introducing the fuel without injection air, an explosion occurs instead of the fuel burning under uniform pressure. For this reason the parts of the "Brons" motor have to be stronger, and

therefore heavier, than if the normal Diesel principle were adopted, a fact which makes this type uneconomical for greater powers than about 30 H.P. The weight of the 24-H.P. "Brons" motor, complete with fly-wheel, shafting, stern tube, bedplate, piping, and fuel supply of 44 gallons, is 8,140 lb., or 339 lb. per H.P. An 8-H.P. motor of the same type, with accessories,

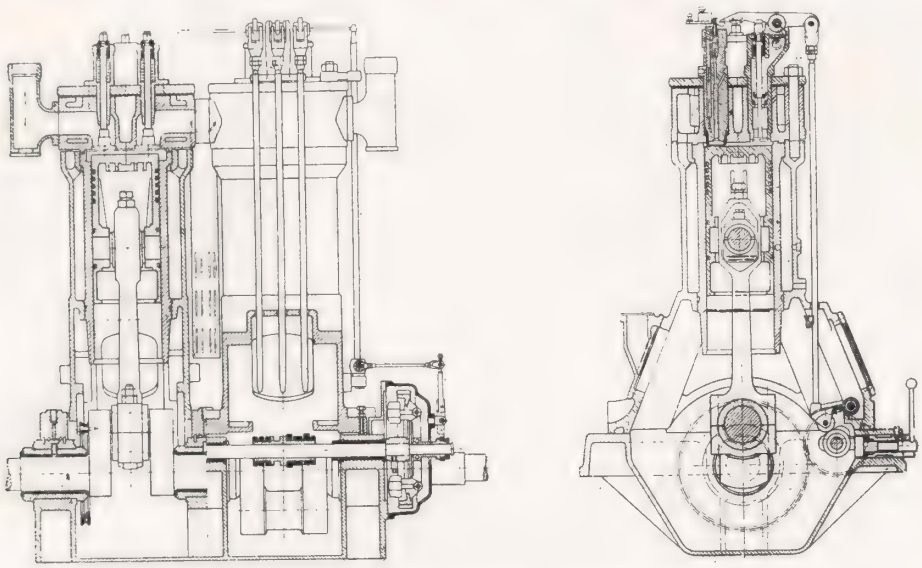


Fig. 1.—24-H.P. Two-cylinder "Brons" Motor.

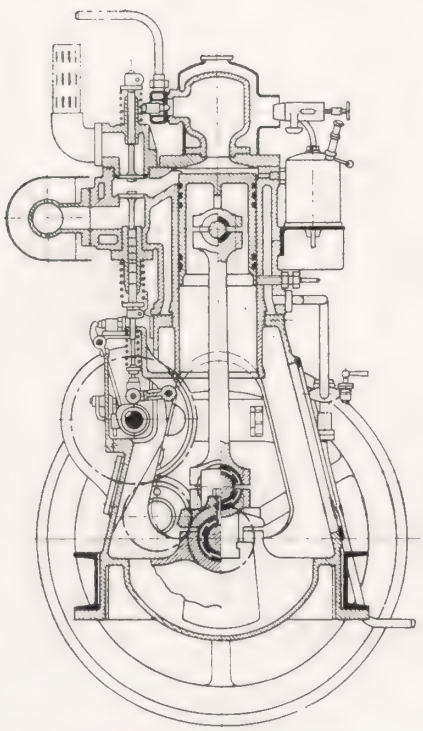


Fig. 2.—8-H.P. "Daevel" Four-cycle Motor.

including a fuel supply of 13 gallons, weighs 3,520 lb., or 440 lb. per H.P. The fuel consumption of the "Brons" motor is from ·53 to ·57 lb. per H.P.-hour.

An earlier type of motor whose use at the present time for fishing boats greatly exceeds that of the more recent "Brons" motor is that in which the ignition of the fuel takes place in a heated chamber or "hot pot" at the top of the cylinder. This type possesses the great advantages of simplicity and cheapness, but, on the other hand, cannot be started until the ignition chamber has been heated by a glow lamp, an operation which occupies from 10 to 15 minutes, while the fuel consumption is greater, being from ·75 to ·84 lb. per H.P.-hour for 6 to 10 H.P. engines. It is essentially a low-pressure motor, while the "Brons" and Diesel motors are high-pressure engines. A good example of a four-stroke cycle two-cylinder motor of this type, made by the Kieler Maschinenbau A.G., Vorm. C. Daevel, of Kiel, is illustrated in Fig. 2. The main particulars in this case are :—

Diameter of cylinders	6½in.
Stroke	8¾in.
Normal revolutions per minute	400
Normal power	8 H.P.
Maximum power	9·45 H.P.

The total weight of the installation, including 31 gallons of fuel oil, is 3,168 lb., or 396 lb. per H.P., normal running.

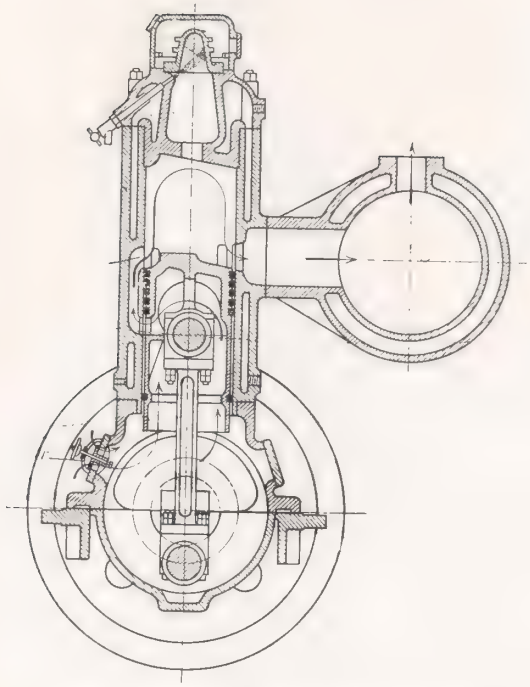


Fig. 3.—8-H.P. "Swiderski" Motor.

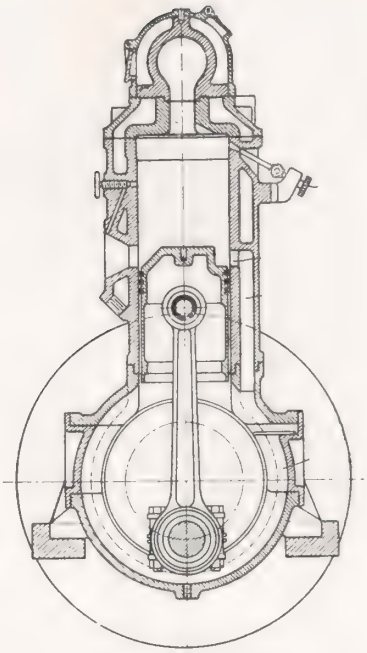


Fig. 4.—8-H.P. "Grade" Motor.

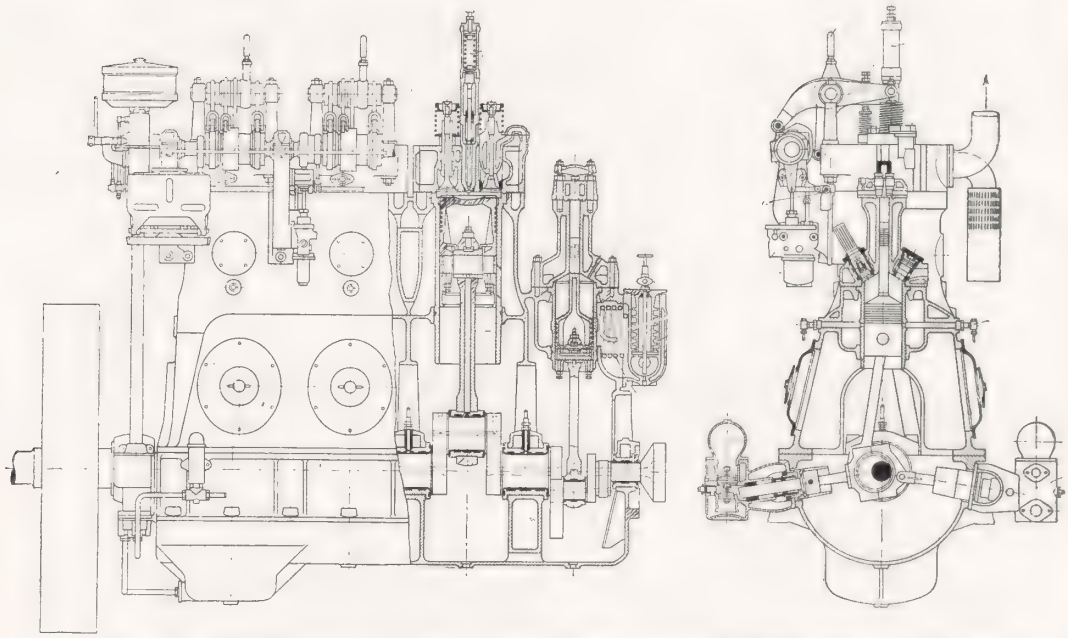


Fig. 5.—75-H.P. Four-cycle Diesel Engine, made by the Gasmotorenfabrik Deutz.

A two-stroke cycle single-cylinder motor of the "hot pot" type is that made by the M.A.G. vorm. P. Swiderski, of Leipzig. The leading dimensions of the Swiderski motor illustrated in Fig. 3 are :—

Cylinder diameter	$7\frac{1}{2}$in.
Stroke	$8\frac{3}{8}$in.
Normal revolutions per minute	375
Normal power	6 H.P.
Maximum power	9 H.P.

Another two-stroke cycle motor which has been adopted with success is that made by the Grade-Motorwerke, Magdeburg, and illustrated in Fig. 4.

With regard to motor installations on larger fishing craft, the author states that not much has yet been accomplished. Messrs. Frerichs and Company, of Osterholz-Scharmbeck, have fitted a 90-H.P. motor to a herring boat, but no information is yet available concerning the experiment. The G.M.F. Deutz have supplied a 75-H.P. four-cycle three-cylinder Diesel engine to a vessel similar in type to a fishing boat for the Biological Institute of Helgoland. This motor, which is illustrated by Fig. 5, has been designed with a view to the subsequent introduction of the type in large fishing boats, and is said to be giving every satisfaction. The main particulars are :—

Cylinder diameter	9½in.
Stroke	12¾in.
Normal revolutions per minute	350
Normal power	75 H.P.

The author then gives plans of the boats in which the motors referred to have been installed, showing the arrangement of the machinery on board. Finally, he discusses the application of oil engines to fishing craft from an economic point of view, and concludes that unquestionably their adoption is worth while in this respect. Even were the extra revenue earned only sufficient to pay for the cost of the engine and fuel, the advantages gained as regards security and improved working conditions for the fishermen and hauling in the nets are sufficient to justify their adoption.

HEAVY-OIL ENGINES FOR MARINE PROPULSION.

By Mr. G. C. Davison.

(*American Society of Naval Architects and Marine Engineers ; November*, 1911.)

ENGINES using crude oil are divided into two general classes—those working on the constant volume or Otto cycle, and those working on the constant pressure or Diesel cycle. The use of the Otto cycle is practically limited to volatile fuels and gases, and up to the present little success has attended attempts to adopt it for heavy oils. On the other hand, the Diesel engine has been a success from the first as a heavy-oil engine. Single-acting Diesel engines are designed on the four-stroke cycle or the two-stroke cycle. It seems to be the general opinion of authorities on the subject that for marine purposes the two-cycle is superior to the four-cycle engine. The superior fuel economy of the four-cycle engine, which amounts to some 8 to 10 per cent., has to be balanced against the even turning moment, the simplicity, the reversibility, the lightness, and the compactness of the two-cycle engine. If, as many engineers confidently expect, the economy of the two-cycle can be made the same as the four-cycle, there will be no room for argument. The vertical two-cycle double-acting engine is the latest development, and several engines of this type have recently been completed and tested. The smallest of these develops 300 H.P. per cylinder, and the largest 2,000 H.P. per cylinder.

The chief problems in the construction of Diesel engines which have had to be considered are those due to the high pressure and temperature in the cylinders, and have largely been solved by development from gas-engine practice. Strong and durable engines of from 150 to 2,500 H.P. can now be constructed with a weight of from 40 to 60 lb. per H.P. and a fuel consumption of from ·40 to ·60 lb. of fuel per horse-power-hour. To show the advantages of heavy-oil engines of the Diesel type as regards economy in fuel, attendance, weight, space, endurance, repairs, reliability, absence of smoke and funnels, cleanliness, readiness for action, and reduced time for loading fuel, Mr. Davison takes the U.S. destroyer *Paul Jones*, comparing the present arrangement of steam propelling machinery in this vessel with a proposed arrangement using oil engines.

The main particulars of the *Paul Jones* are as follow :—

Length	250ft.
Beam	23ft. 6in.
Draught (mean)	7ft.
Displacement (sea-going trim)	470 tons
Coefficient of fineness	·41

The vessel is provided with four Thornycroft water-tube boilers, designed for a working pressure of 300 lb. per square inch and supplying steam to two four-cylinder vertical triple-expansion engines each capable of developing about 4,000 H.P. The arrangement of the machinery is as shown in Fig. 1.

A comparative arrangement, with three oil engines of 2,500 shaft horse-power each, is shown in Fig. 2. The engine which has been selected is one of the Nuremburg type now building, and

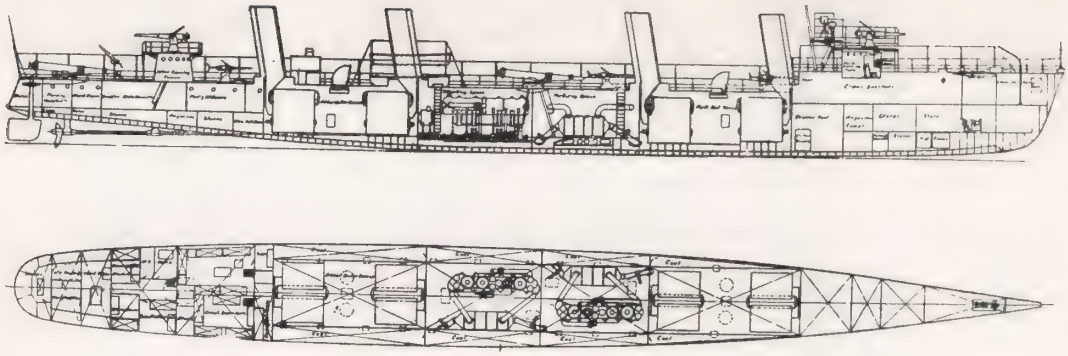


Fig. 1.—The "Paul Jones," with present Steam Plant.

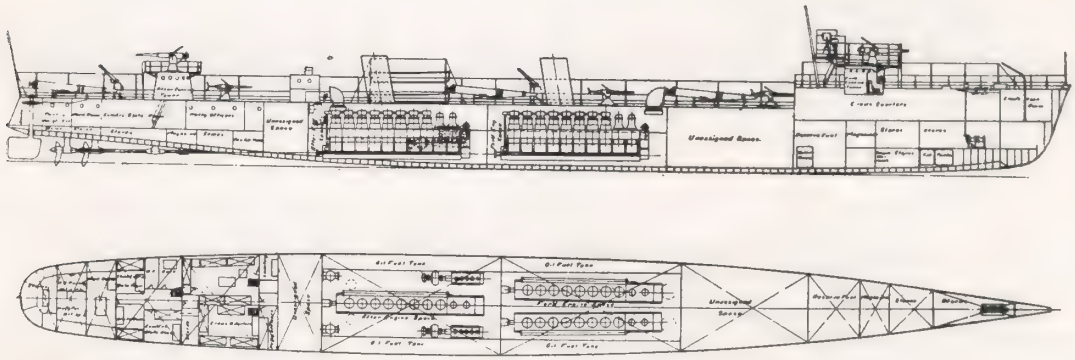


Fig. 2.—The "Paul Jones," with Heavy-Oil Engines.

of which complete plans and data are available. It is of the eight-cylinder, single-acting, and two-cycle type, with general dimensions as follow :—

Diameter of cylinders	18·9in.
Stroke	19·7in.
Revolutions per minute	350
Brake horse-power	2,500
Length overall	34ft.
Breadth	5ft.
Height	10ft.
Weight, complete	47 tons

The advantages of the oil-engine arrangement are tabulated in Table I., and are so great that the author considers it will not be long before high-powered vessels equipped with internal-combustion engines will become quite familiar objects.

TABLE I.

Item.	Steam.	Oil.
Indicated horse-power	7,700	8,300
Shaft horse-power	7,000	7,200
Weight of main motive power machinery, lb.	449,522	317,000
Weight per shaft horse-power, lb.	64	44
Cruising radius at 16 knots with 180 tons fuel, miles	1,700	10,000
Pounds fuel per horse-power at 16 knots	3·0	0·60
Cruising radius at 28 knots with 180 tons of fuel	630	2,950
Fuel per horse-power at 28 knots, lb.	2·34	0·50
Engineering staff required	54	21
Fuel consumed per annum 20,000 knots cruising, tons	2,100	360
Fuel consumed in port, tons	300	300
Annual total fuel consumption, tons	2,400	660
Cost of fuel on Pacific Coast	$19,200	$4,620
Pay of engineering staff per year	$22,500	$9,600
Repairs, average annual	$10,000	$2,000

Mr. IVAR KNUDSEN,
Technical Director of Messrs. Burmeister & Wain, Copenhagen.

Ivar Knudsen

RESULTS OF TRIALS OF THE DIESEL-ENGINED SEA-GOING VESSEL "SELANDIA."*

By Mr. I. KNUDSEN.

(*Institution of Naval Architects; March*, 1912.)

THE *Selandia* is the first to be completed of two vessels for which the order was placed by the East Asiatic Company, of Copenhagen, with Messrs. Burmeister & Wain on the 5th December, 1910. The general arrangement of these two ships is shown in Fig. 2. Their dimensions are :—Length B.P., 370ft.; breadth, 53ft.; and depth moulded to upper deck, 30ft. The propelling machinery consists in each case of two sets of eight-cylinder Diesel motors driving twin screws. The engines work on the four-stroke cycle system, and develop a total indicated horse-

* This vessel was fully described and illustrated in No. 24, Vol. VI. of *The Shipbuilder*.

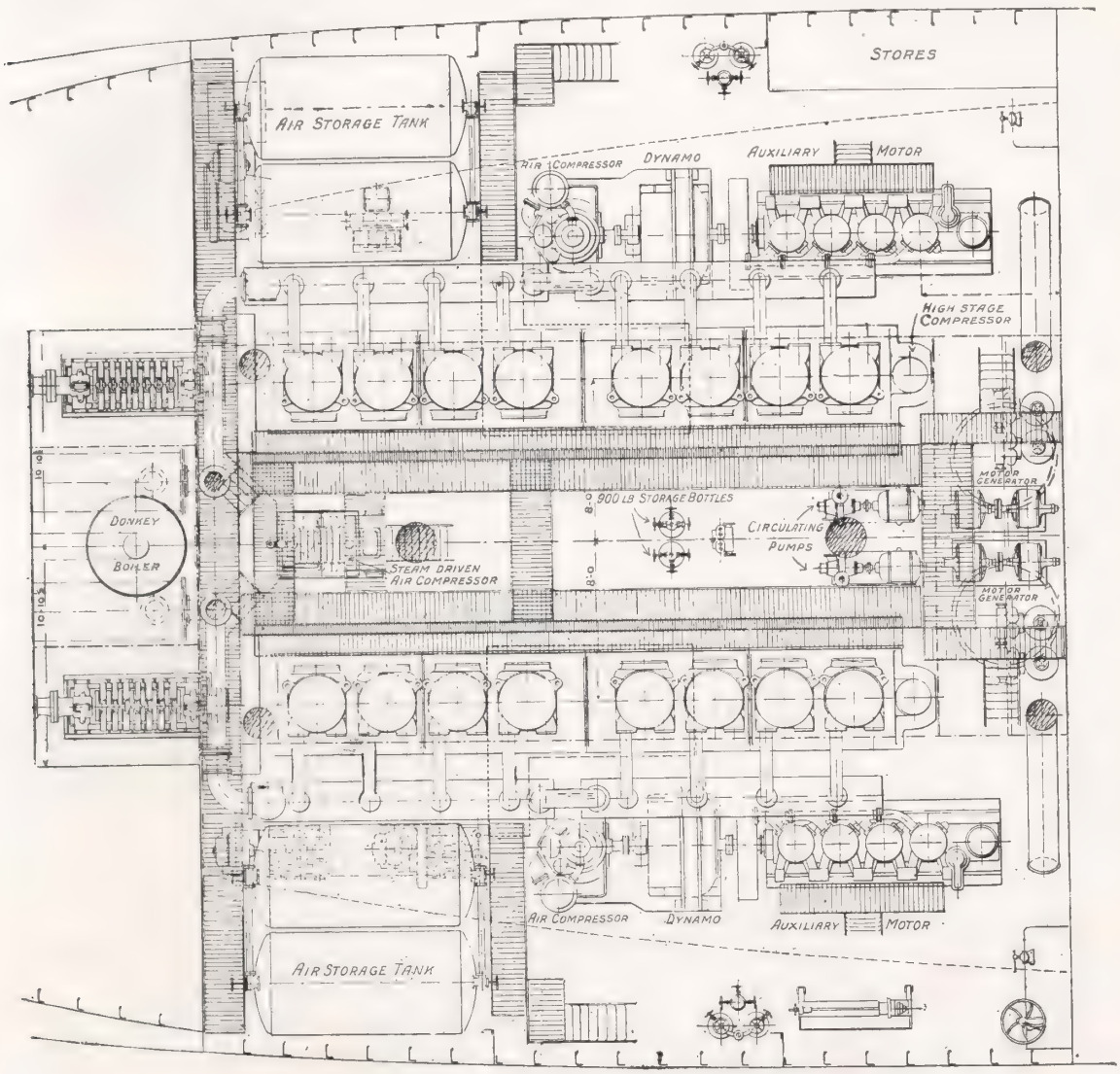


Fig. 1.—Plan of the "Selandia's" Engine Room.

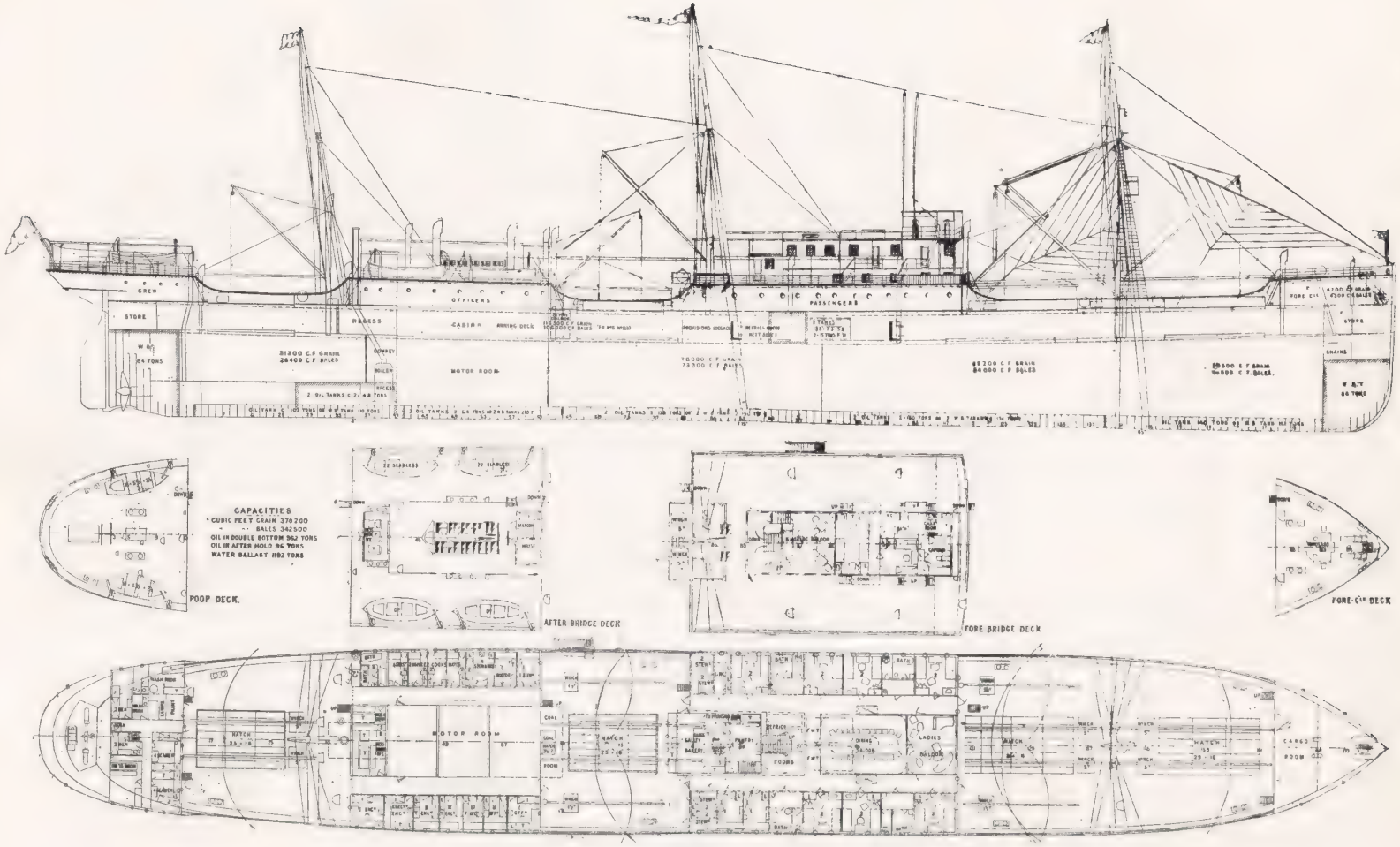


Fig. 2.—General Arrangement of the "Selandia."

power of about 2,500 at the normal speed of 140 revolutions per minute. There are also two auxiliary Diesel motors of 250 I.H.P. each, working at about 230 revolutions per minute, and each driving direct a dynamo and a three-stage air compressor capable of compressing air to 300lb. per square inch. The arrangement of the machinery is clearly shown on the plan of the engine room, Fig. 2. The starting of the engine takes place in either direction by means of compressed air, which is introduced into the cylinders through the starting valves at a pressure of 300 lb. per square inch. The reversing from full speed ahead to full speed astern can be carried out in less than 20 seconds. The speed of each engine at sea is controlled by an Aspinall's governor.

A small flywheel 6ft. 6in. in diameter is provided on each shaft, and has toothed-wheel gearing on the periphery, which is driven by an electric motor when it is required to turn the engine.

An air compressor is provided on each main engine at the forward end, capable of compressing air from 300 lb. to 900 lb. per square inch, which is the pressure required for injecting fuel oil into the cylinders. There is also a small steam-driven compressor, which can compress air to 900 lb. per square inch. The steam for driving this compressor and for heating and fire-extinguishing purposes is provided by a donkey boiler placed at the after end of the engine room.

Most of the engine room auxiliaries and all the deck machinery are electrically driven. The steering gear is of the Hele-Shaw Martineau hydraulic-electric type.

The fuel oil is stored in the double bottom of the vessel and in two tanks (which are provided with heating coils) at the sides of the shaft tunnels.

After making three short trial trips in the Sound, the *Selandia* left Copenhagen on the 22nd February for Aalborg, and from there the voyage was continued to London. Throughout this voyage the machinery worked excellently. Records were taken of the consumption of fuel oil, which was found to be ·363 lb. of oil per I.H.P. developed by the main engines per hour, this figure including all the fuel oil used by the auxiliary motors.

GAS POWER FOR SHIP PROPULSION.

By Mr. A. C. Holzapfel.

(*Institution of Naval Architects; March*, 1912.)

AFTER dealing with earlier attempts to use gas engines for ship propulsion, the author describes the small vessel *Holzapfel I.*, which was built to test the adaptation and installation of gas power for propelling purposes on board a sea-going ship, and in particular to test the use of the Föttinger hydraulic transformer in conjunction with gas engines. The *Holzapfel I.* has the following dimensions:—Length, 120ft.; beam, 22ft.; depth, 11ft. 6in.; and she carries 370 tons deadweight on Lloyd's summer freeboard. Her general arrangement is shown in Fig. 1.

The gas plant was built and supplied by the Power-Gas Corporation, of Stockton-on-Tees, and works with anthracite coal. It consists of two producers 3ft. 6in. square and 6ft. 6in. high, and two scrubbers 13ft. high, and 6ft. 6in. diameter. These are placed on a platform 3ft. 9in. high, and they form part of a closed-in gas space, which is accessible by a gas-tight door from the engine room. The vertical gas engine was made by Messrs. E. S. Hindley & Sons, of Bourton, Dorset. It has six cylinders, 10¾in. diameter and 10in. stroke, and is designed to run at 450 revolutions per minute. The engine has forced lubrication, low tension magneto, and also Lodge ignition. The Föttinger transformer drives the propeller at about 120 revolutions per minute. Experiments on the test plate before delivery showed the loss of power in the transformer to be from 11¾ to 18 per cent.

The vessel had her trials at the end of April, 1910, and has since made about a dozen laden voyages. The consumption of anthracite coal during these runs varied between 25 and 35 cwt. per day of 24 hours, which is about one-half that of steamers of similar power. During this period several faults manifested themselves in connection with the machinery and particularly with the gas plant; but the author is of opinion that, with the experience gained and the alterations which have been made, the vessel will be able to do her work in a satisfactory manner.

The main purpose of the owners in building the *Holzapfel I.* was to evolve from the experience with this vessel a design for ships of large size to be driven by gas engines. In a large vessel, it would, for economic reasons, be absolutely essential to use bituminous instead of anthracite coal. Figs. 2 and 3 show the proposed arrangement of a bituminous gas plant of 800 H.P. on board ship. In this arrangement the producers

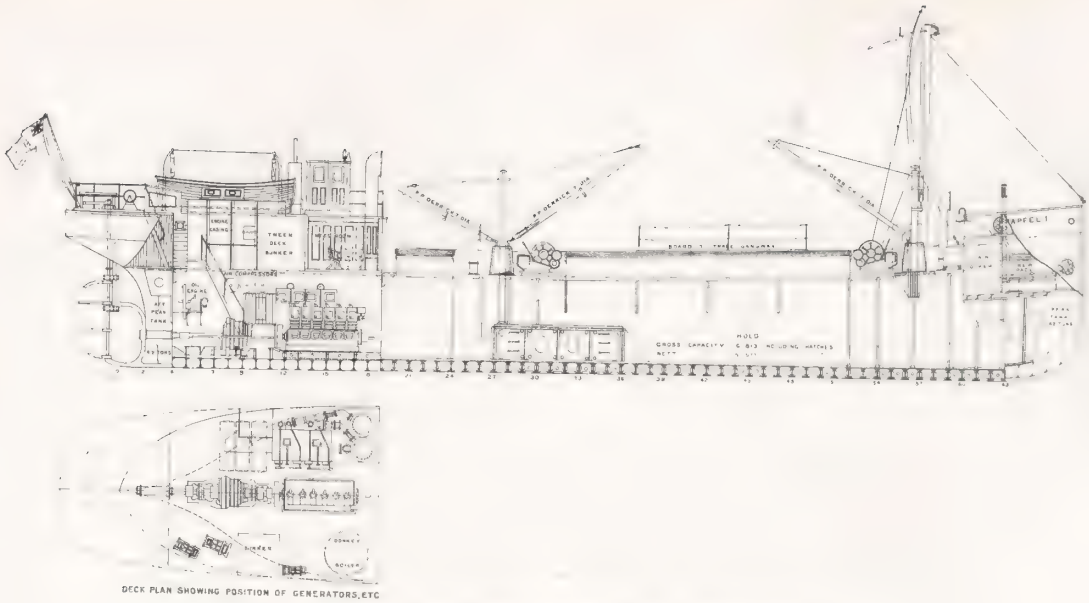


Fig. 1.—General Arrangement of the "Holzapfel I."

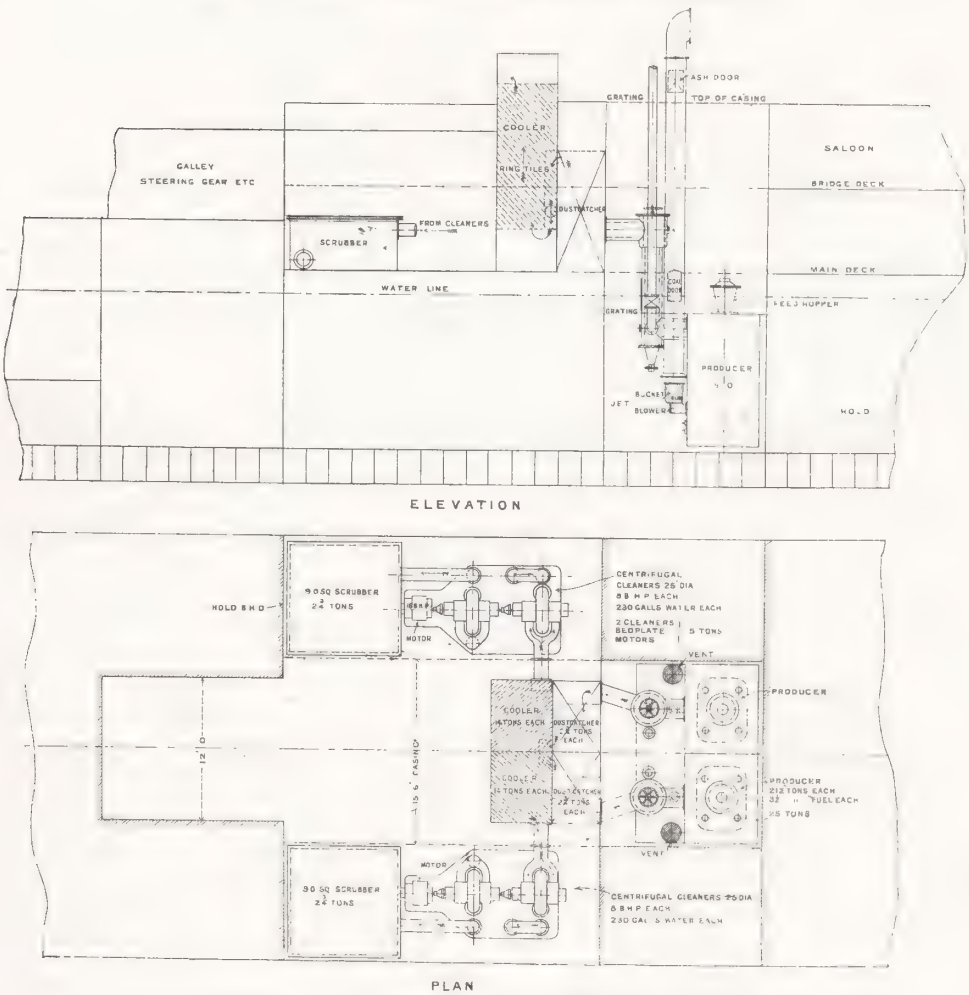
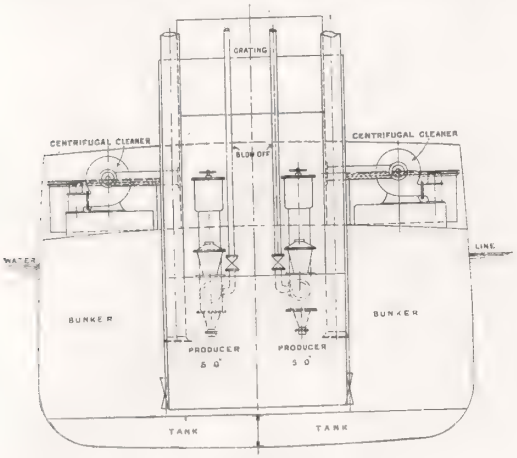


Fig. 2.—Arrangement of 800-B.H.P. Plant.

would stand on the tank top in a separate compartment, the coal being fed from a 'tween deck bunker. From the producers the gas would pass into the coolers, which would be about 15ft. high and 3ft. 9in. diameter, and placed on the main



CROSS SECTION

Fig. 3.—800=B.H.P. Plant.

deck so that cooling water could be discharged overboard by gravitation. From the coolers the gas would be drawn by centrifugal tar extractors and forced under pressure through the dry scrubbers to the engines.

Coming to the main engines, the author considers that the twin-screw system should be adopted on account of the additional security afforded. In order to reduce the vibration to a minimum, engines of six or more cylinders are desirable. Two types of vertical engines are at present being built, those with tandem cylinders and those with a single row of cylinders; the former have the advantage in saving space, while the latter are, perhaps, more accessible for repairs. The stroke of gas engines is generally limited to a piston speed of about 800ft. per minute; and engines such as have been considered suitable, and of the size not requiring water-cooling of pistons, will run at about 250 to 320 revolutions per minute.

As regards the Föttinger transformer, about 85 to 90 per cent. of the original brake horsepower of a prime mover of 800 B.H.P. can be transmitted to the propeller shaft if the revolutions are reduced from about 270 of the prime mover to 100 of the propeller. The weight of the transformer and water would be about 13 tons.

In conclusion, the author points out that the unlimited coal supply and the cheap price of this fuel in Great Britain, as compared with liquid fuel, makes the introduction of marine gas power particularly desirable in this country, not only from economic but also from national considerations.

SOME ASPECTS OF DIESEL ENGINE DESIGN.

By Mr. D. M. Shannon.

(Institution of Engineers and Shipbuilders in Scotland, April, 1912.)

THE oil engine is still in its infancy, and the greatest care is required in its design and manufacture, due in the first instance to the high pressures which have to be dealt with, and, secondly, to the small volume occupied by the power medium. In an oil engine one cubic inch of oil for power purposes is, on the average, equivalent to 5,500 cubic inches of steam used in a steam engine. Hence, the measuring and burning arrangements must be made with great accuracy, as any slight inaccuracies may cause large variations in the power and efficiency of the machine. Oil engine workers recognise that the piston engine is only a temporary solution of the problem and that a reversion to the turbine principle will be the natural outcome of the inherent limitations and disadvantages of the reciprocating form of motor. Facts, however, seem to point strongly against the success of the internal-combustion turbine working any of the present cycles; and until a new cycle or a new heat carrier can be found, there is little hope of much progress being made in this direction.

The author then considers the question as to how large the Diesel engine can be made for ship installations. Considering that 2,500 H.P. has been put into one cylinder, he assumes that it is possible to put 2,000 H.P. into one single-acting two-cycle cylinder, or 4,000 H.P. into one double-acting two-cycle cylinder. By combining twelve single-acting two-cycle cylinders as in Fig. 1, or six double-acting cylinders as in Fig. 2, then a total of 24,000 H.P. is reached for one shaft, and if three shafts are used 72,000 B.H.P. could be obtained in one ship.

The pistons of these motors would be about 45in. diameter and the stroke 60in. The crankshaft would be 27in. diameter stressed to 11,000

lb. per square inch. The overall dimensions in the two cases would be as shown in the illustrations, the two halves of the single-acting motor being in separate engine rooms. The large amount of space required for the single-acting engine seems to show that for really large powers the two-cycle double-acting type is the only one that can be seriously considered.

One of the greatest advantages of the Diesel engine is that, however small the quantity of combustible matter injected into the cylinder may be, it is certain to burn. This fact makes the Diesel cycle eminently suitable for dealing with gaseous fuels, especially for reversing engines, and the author thinks that this is the correct line to work along in developing gas plants for ship propulsion.

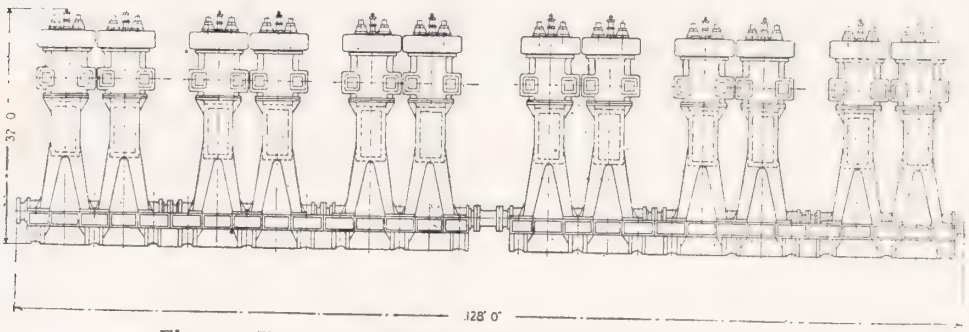


Fig. 1.—Two-cycle Single-acting Engine, 24,000 B.H.P.

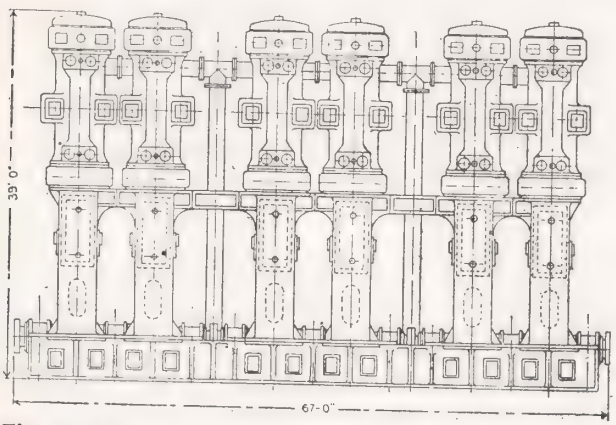


Fig. 2.—Two-cycle Double-acting Engine, 24,000 B.H.P.

With regard to the design of crankshafts for Diesel engines, Tables I. and II. contain a fairly complete analysis of the twisting moments in the case of all engines of the four and two-cycle single-acting types which have actually been built or proposed. In drawing up these tables the effect of compressor and scavenging cranks has been neglected, and the constants, although quite good for preliminary design work, are given rather for purposes of comparison. Three sets of constants are given in each case, the first neglecting inertia and the second and third allowing for inertias of 100 and 200 lb. per square inch of piston area respectively. The indicator cards used for calculating the twisting moment curves are shown in Fig. 3, the connecting rod being assumed to be 4·5 cranks long.

In columns 5, 7 and 9, Tables I. and II., the ratio R of maximum twisting moment in the shaft to mean twisting moment is given, and the number in brackets immediately under denotes the number of cylinders giving this maximum. In columns 6, 8 and 10, R_1 is the ratio of the twisting moment for the total number of cylinders to the mean twisting moment. Ratios R will, therefore, be used for crankshaft design, and ratios R_1 for tunnel-shaft design. It will be noticed that, in a number of cases, inertia makes very little difference in R_1, that in four-cycle work inertia cannot be neglected, and that in some extreme cases there is an increase of 60 per cent. in the twisting moment due to inertia. In two-cycle engines the effect of inertia in nearly every case reduces the twisting moment and it is quite safe to neglect it altogether.

The following formula is proposed by the author for determining the diameter of the crankshaft :—

Let d = diameter of shaft in inches.

D = diameter of cylinder in inches.

f = stress in lb. per square inch.

P = mean pressure in cylinder lb.

Table I.—Four-cycle Single-acting Engine.

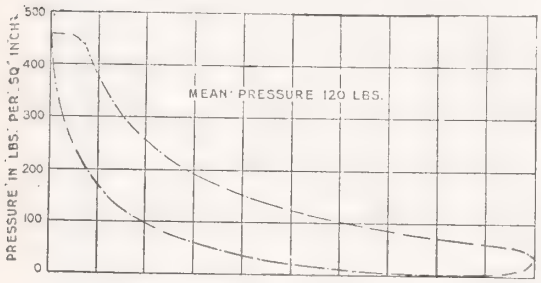
1	2	3	4	5	6	7	8	9	10	11	12	13
				VALUES OF R & R_1.						ABSOLUTE HEIGHT OF MAX. T.M. CURVE		
				Inertia—0.		Inertia—100lb.		Inertia—200lb.				
No. of Cranks	Crank Angles.	Order of Firing.	CRANK SHAFT.	R.	R_1.	R.	R_1.	R.	R_1.	0.	100lb.	200lb.
1	0	1		14·2 (1)	14·2	10·8 (1)	10·8	9·1 (1)	9·1	2·46	1·88	1·58
2	360°	1,2		7·1 (2)	7·1	5·4 (1)	4·6	7·1 (2)	7·1	2·46	1·88	2·46
2	180°	1,2		7·1 (2)	7·1	5·4 (1)	4·5	6·8 (2)	6·8	2·46	1·88	2·37
3	120°	1,2,3		4·7 (3)	4·7	4·2 (2)	3·78	5·15 (2)	4·3	2·46	2·19	2·69
4	180°	1,2,4,3		3·5 (4)	3·5	2·7 (1)	2·34	5·15 (4)	5·15	2·46	1·88	3·58
4	90°	1,2,4,3		4·3 (4)	4·3	4·15 (4)	4·15	4·4 (3)	4·25	2·98	2·88	3·08
5	72°	1,3,5,4,2		2·86 (5)	2·86	3·35 (4)	2·8	3·9 (4)	2·8	2·49	2·92	3·4
6	120°	1,2,3,6,5,4		2·55 (5)	2·42	2·3 (5)	1·44	2·45 (2)	2·02	2·65	2·4	2·55
6	120°	1,5,3,6,2,4	SAME AS ABOVE.	2·55 (5)	2·42	2·17 (5)	1·44	2·6 (2)	2·02	2·65	2·26	2·69
8	90	1,5,2,6,4,8,3,7		2·16 (5)	1·74	2·54 (5)	1·61	3·4 (5)	1·63	3	3·53	4·73
8	90	1,6,2,4,8,3,7,5		2·16 (4)	1·74	2·4 (5)	1·61	2·9 (6)	1·63	3	3·33	4·02
12	60	1,2,3,6,5,4, 7,8,9,12,11,10	TWO SIX-CYL. ENGINES ARRANGED AT 60°.	1·74 (7)	1·17	1·77 (8)	1·27	2·01 (10)	1·21	3·62	3·68	4·18
12	60°	1,5,3,6,2,4, 7,11,9,12,8,10	SAME AS ABOVE.	1·77 (7)	1·17	1·8 (10)	1·27	2·06 (10)	1·21	3·68	3·74	4·29

Table II.—Two-cycle Single-acting Engine.

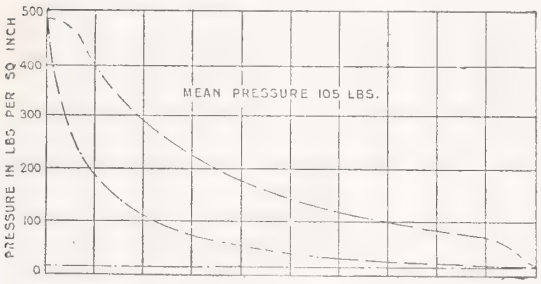
1	2	3	4	5	6	7	8	9	10	11	12	13
				VALUES OF R. & R_1.						ABSOLUTE HEIGHT OF MAX. T M. CURVE.		
				Inertia—0.		Inertia—100lb.		Inertia—200lb.				
No. of Cranks.	Crank Angles.	Order of Firing.	CRANK SHAFT.	R.	R_1.	R.	R_1.	R.	R_1.	0.	100 lb.	200 lb.
1	0	1		7·6 (1)	7·6	5·8 (1)	5·8	5 (1)	5	2·7	2·06	1·78
2	180°	1,2		3·8 (1)	3·64	2·9 (1)	2·25	2·5 (1)	2·25	2·7	2·06	1·78
3	120	1,2,3		2·7 (2)	2·3	2·38 (2)	1·83	2·14 (2)	1·37	2·87	2·52	2·27
4	90°	1,4,2,3		2·22 (3)	1·68	1·93 (3)	1·65	2·1 (3)	1·62	3·16	2·74	3·02
5	72°	1,5,2,3,4		1·91 (3)	1·31	1·64 (4)	1·29	1·93 (4)	1·28	3·38	2·9	3·42
5	72°	1,3,5,2,4		1·91 (3)	1·31	1·68 (4)	1·29	1·96 (4)	1·28	3·38	2·96	3·47
6	60°	1,6,2,4,3,5		1·7 (5)	1·12	1·62 (4)	1·17	1·62 (5)	1·17	3·7	3·44	3·45
6	60°	1,4,5,2,3,6		1·71 (5)	1·12	1·48 (5)	1·17	1·62 (5)	1·17	3·64	2·94	3·45
8	45°	1,7,5,4,2,8,6,3		1·64 (5)	1·03	1·24 (7)	1·02	1·41 (6)	1·01	4·66	3·78	4
8	45	1,8,6,4,2,7,5,3		1·48 (7)	1·03	1·32 (5)	1·02	1·43 (7)	1·01	4·2	3·74	4·05

P_m = Maximum pressure in cylinder at instant of maximum twisting moment.
= 420 lb. per square inch approximate.

n = number of working cylinders.



Four-cycle Card.



Two-cycle Card.

Fig. 3.—Indicator Diagrams.

$$R_m = \frac{\text{maximum twisting-moment}}{\text{minimum twisting-moment.}}$$

$$a = \frac{\text{distance between centres of bearings}}{\text{diameter of cylinder.}}$$

= 2 for most engines.

$$b = \frac{\text{piston stroke}}{\text{diameter of cylinder.}}$$

= 1·2 for high-speed engines and 1·4 for low-speed engines.

Then x = $PbnR_m$ for 4-cycle single-acting engines.
= 2 $PbnR_m$ for 2-cycle single-acting engines.
= 4 $PbnR_m$ for 2-cycle double-acting engines.

$$y = \frac{P_m\ a}{x}$$

Then

$$d = D\sqrt[3]{\frac{x}{f}\left\{\cdot 35\,y + \cdot 65\sqrt{y^2 + \cdot 4}\right\}}$$

$$\text{or } f = \frac{D^3}{d^3}x\left\{\cdot 35\ y + \cdot 65\sqrt{y^2 + \cdot 4}\right\}$$

These formulæ, although rather complicated, take into account all the variables and constants. x and y can be worked out for a particular type of engine and tabulated for use in design work. It is deduced from the usual equation—

$$M_e = \cdot 35\ M_b + \cdot 65\sqrt{M_b^2 + M_t^2}$$

the amount of bending allowed for being equal to $\frac{WL}{8}$ where W = piston load, and L = distance between centres of bearings.

The author also deals very fully with the design of main bearing and connecting-rod caps and brasses, with valve gear diagrams, and with the design of the air compressors and scavenging air plant, numerous formulæ and illustrations being given in the paper regarding these matters.

Trials of Warships.

THE U.S.S. "UTAH."

By Lieutenant-Commander W. R. White, U.S.N.

(*American Society of Naval Engineers; August*, 1911.)

THE *Utah*, built by the New York Shipbuilding Co., Camden, N.J., is one of two ships of this class authorized by Act of Congress on the 13th May, 1908. The contract was signed on the 24th November, 1908; the keel was laid on the 22nd February, 1909; and the hull was launched on the 23rd December, 1909. The official trials took place from the 26th June to the 2nd July, 1911. The contract price for the construction of the hull and machinery was $3,946,000, of which sum $2,846,000 was allotted to the hull and $1,100,000 to the propelling machinery and auxiliaries. The dimensions of the hull are:—Length B.P. on waterline, 510ft. 0in.; breadth extreme, 88ft. 2⅝in.; depth moulded to main deck, 44ft. 6in. The propelling machinery consists of Parsons turbines arranged on four shafts, six turbines being provided for going ahead and four for going astern, arranged as shown in Fig. 1. The four propellers are each the same size, and are solid castings of manganese bronze with three blades, all turning outboard for the ahead motion. The diameter of each propeller is 9ft. 2in., pitch 8ft. 6in., and developed area 41 sq. ft. There is one main condenser in each engine room, located at the after end, each condenser having a cooling surface of 15,235 sq. ft. The boiler installation consists of 12 Babcock & Wilcox water-tube boilers designed for a working pressure of 200 lb. per square inch, and placed four each in three watertight compartments. The total heating surface is 64,234 sq. ft., and grate surface 1,428 sq. ft. The total weight of all machinery is 2,064 tons. The results of the standardization runs over the measured mile course at Rockland, Maine, are given in Table I., and the results of other official trials in Table II. The paper also contains very complete information with regard to the size, expansion stages, and blading of the turbines, and full particulars of the boilers, condensers, and all auxiliary machinery

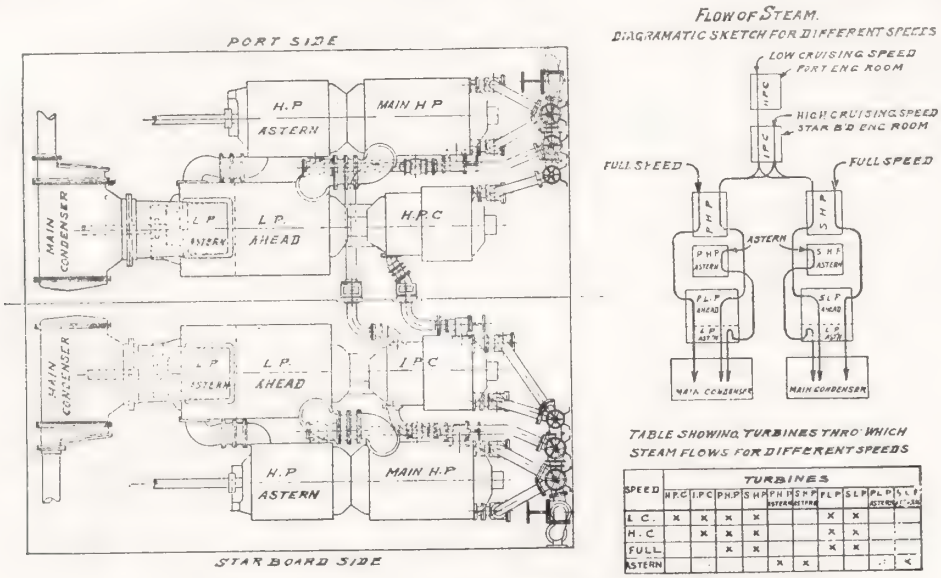


Fig. 1.—U.S.S. "Utah": Arrangement of Turbine Machinery.

TABLE I.—STANDARDIZATION RUNS.

Draught	Forward, 27ft. 7½in.; aft., 28ft. 1¼in.; mean, 27ft. 10⅜in.					
Displacement	21,268 tons.					
Speed in knots	10·56	12·02	16·80	19·05	20·63	21·29
Revolutions per min.	151	172	244	278	312	330
Shaft H.P. of all turbines	2,595	3,823	11,150	16,486	23,588	28,842
Effective H.P. (without appendages) ...	1,360	2,080	6,200	9,050	12,650	15,000
Propulsive coefficient	...	·55	·554	·547	·532	·518

TABLE II.—OFFICIAL TRIALS.

	24-hour 12 knots.	24-hour 19 knots.	4-hour full power.	2-hour oil & coal.
Mean draught	27′ 10 9/16″	27′ 10½″	27′ 10 3/16″	27′ 9 13/16″
Trim by the stern in inches	4⅞	3⅛	9 13/16	12⅜
Displacement in tons	21,299	21,292	21,284	21,261
Weather	Calm, fog	Light breeze	Light breeze	Light breeze
Sea	Smooth	Smooth	Smooth	Smooth
Number of boilers in use	6	12	12	12
Speed of ship in knots per hour	12·02	19·23	21·04	21·04
Slip of propeller, per cent.	18·27	18·54	22·25	22·54
Main steam pressure at boilers, maximum, lb. sq. in. ...	215	223	225	222
Do. do. do. mean, do. ...	199	208	213	205
Vacuum in condensers, mean, inches	29·14	28·55	28·15	28·0
Mean revolutions of all shafts	173	282	323	323
Total I.H.P. of all engine auxiliaries	542	838	1,100	1,097
Shaft H.P. of all turbines	3,938	17,150	27,036	27,054
Total H.P.	4,480	17,988	28,136	28,151
Effective H.P. for bare hull	2,200	9,750	15,900	15,900
Steam consumption, lb. per hour per S.H.P., main turbines only	25·01	15·30	15·03	..
Steam consumption, lb. per hour per H.P. of main turbines and engine auxiliaries	21·98	14·59	...	...
Heat value of coal used, B.T.U. per lb.	14,263	14,874	14,717	14,795
Heat value of fuel oil used, B.T.U. per lb.	...	...	...	19,421
Coal burnt per hour, lb.	11,085	26,393	43,512	31,024
Fuel oil burnt per hour, lb.	...	...	...	8,617
Coal per hour equivalent to oil burnt	...	...	...	11,312
Total coal per hour and coal equivalent of oil	...	...	...	42,336
Pounds of coal per hour per S.H.P., main engines ...	2·815	1·539	1·609	1·147
Do. do. do. all machinery ...	2·474	1·467	1·546	1·102

DESCRIPTION AND TRIALS OF THE U.S. TORPEDO-BOAT DESTROYERS "WARRINGTON" AND "MAYRANT."

By Mr. W. B. Robins.

(*American Society of Naval Engineers; November*, 1911.)

THESE vessels contain the first installations of Zoelly turbines for naval propulsion in the U.S. Navy. The contract for these vessels, signed on the 1st October, 1908, called for delivery within 24 months, the contractors being the William Cramp & Sons Ship and Engine Building Co., Philadelphia. The *Warrington* was delivered in March, 1911, and the *Mayrant* in July, 1911. The contract price for each vessel was $664,000, of which $389,000 was allotted for machinery. The principal dimensions of each vessel are :—

Length B.P. on load waterline	289ft. 0in.
Breadth moulded	26ft. 4½in.
Depth moulded	16ft. 4⁷⁄₁₆in.
Draught	8ft. 4in.
Displacement	742 tons.

The turbine propelling machinery consists of two Zoelly compound impulse turbines arranged on two shafts, each having a backing turbine in the after end. Each turbine is composed of two essential parts, the fixed part, or casing, and the moving part, or rotor. In the casing are mounted the fixed diaphragm plates and expanding guide blades and also the intermediate guide blades, and on the rotor are mounted the discs and drum carrying the moving blades. The rotor blades and the intermediate guide blades in the casings are of drawn brass. The top clearance is $\frac{5}{64}$in., and the longitudinal clearance $\frac{3}{16}$in.

There are four White-Forster water-tube oil-burning boilers arranged in two fire-rooms, having a total heating surface of 18,000 square feet, and designed for a working pressure of 250 lb. per square inch. The weight of one boiler complete with water at working level is 24·3 tons. The Schütte and Koerting system of oil-burning is used, with independent air registers of the Babcock & Wilcox type. Each boiler is provided with eleven oil burners.

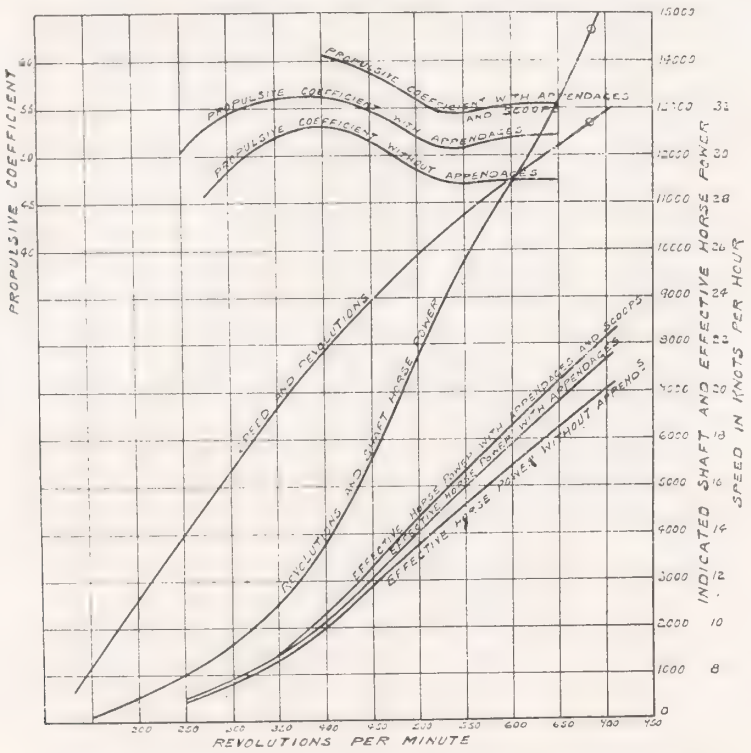

PROPULSIVE COEFFICIENT
PROPULSIVE COEFFICIENT WITH APPENDAGES AND SCOOPS
PROPULSIVE COEFFICIENT WITH APPENDAGES
PROPULSIVE COEFFICIENT WITHOUT APPENDAGES
SPEED IN KNOTS
INDICATED SHAFT AND EFFECTIVE HORSE POWER
SPEED IN KNOTS PER HOUR
REVOLUTIONS PER MINUTE

Fig. 1.—Standardization Trials, U.S.S. "Warrington."

The results of the standardization trials of the two vessels are shown in Figs. 1 and 2. The results of the four and ten-hour trials are given in Table I.

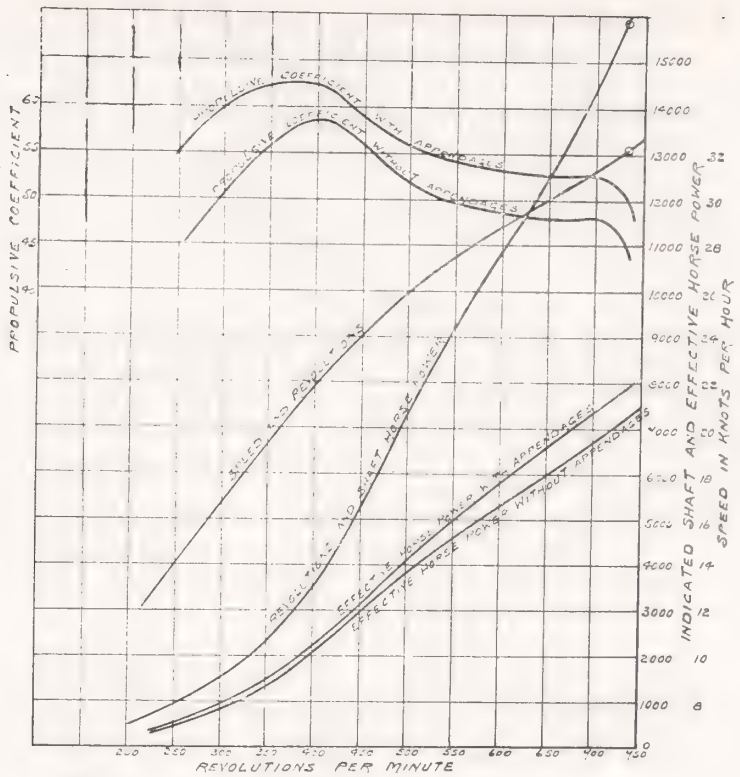

PROPULSIVE COEFFICIENT
INDICATED SHAFT AND EFFECTIVE HORSE POWER
SPEED IN KNOTS PER HOUR
REVOLUTIONS PER MINUTE
PROPULSIVE COEFFICIENT WITH APPENDAGES
PROPULSIVE COEFFICIENT WITHOUT APPENDAGES
SKIED AND REVOLUTIONS
INDICATED SHAFT HORSE POWER
EFFECTIVE HORSE POWER WITH APPENDAGES
EFFECTIVE HORSE POWER WITHOUT APPENDAGES

Fig. 2.—Standardization Trials, U.S.S. "Mayrant."

TABLE I.—OFFICIAL TRIALS, U.S.S. "WARRINGTON" AND "MAYRANT"

	Four-hour full-speed Trials.		Ten-hour Trials. 25 knots.		Ten-hour Trials. 16 knots.	
	Warrington.	*Mayrant.*	*Warrington.*	*Mayrant.*	*Warrington.*	*Mayrant.*
Displacement, tons	729	734	735	726	727	746
Steam pressure at boilers, lb.	246	246	246	245	248	249
Vacuum in condenser, in.	29	28·5	29	28·3	29·2	28·4
Revs. per minute	641	659	502	488	292	292
Speed of ship, knots per hour	30·12	30·22	25·93	25·44	16·27	16·27
Slip of propellers, per cent.	22·89	24·87	15·30	14·54	8·49	8·78
Shaft horse-power	12,846	13,140	7,803	6,840	1,607	1,445

DESCRIPTION AND TRIALS OF THE U.S. TORPEDO-BOAT DESTROYER "PATTERSON."

By Mr. W. B. Robins.

(*American Society of Naval Engineers; November*, 1911.)

THE contract for this vessel was signed on the 14th June, 1909, and called for delivery within 24 months, the contractors being the William Cramp & Son Ship and Engine Building Company, Philadelphia, Pa. The *Patterson* was launched on the 29th April, 1911, and was delivered on the 7th October, 1911. The contract price was $637,000, of which $367,000 was allotted for machinery. The leading particulars of the hull are the same as in the case of the *Warrington* and *Mayrant*.

The propelling machinery consists of Parsons turbines, the usual five-turbine three-shaft arrangement being adopted in conjunction with White-Forster oil-burning water-tube boilers.

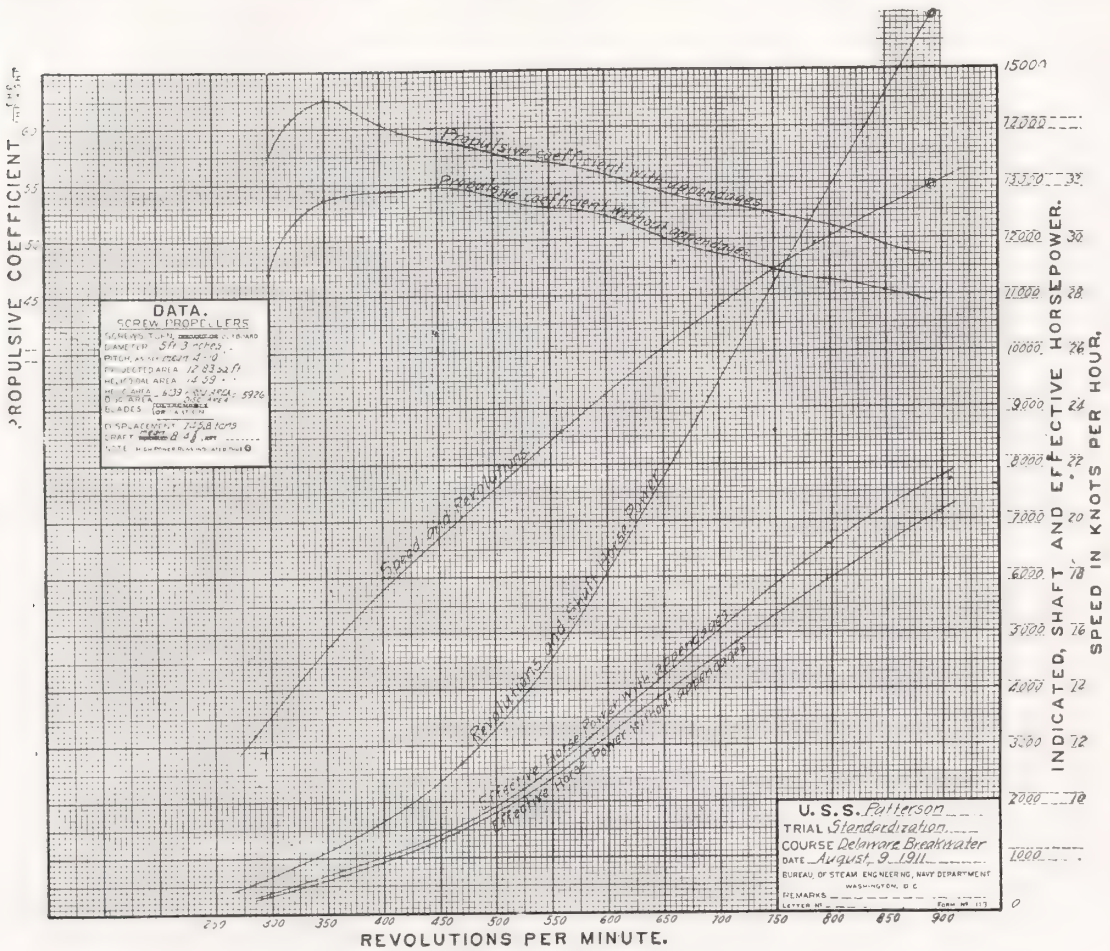


Fig. 1.—Standardization Trials, U.S.S. "Patterson."

The results of the standardization trials are shown in Fig. 1, and of the 4-hour trials in Table I.

TABLE I.—OFFICIAL 4-HOUR TRIALS, U.S.S. "PATTERSON."

Mean Draught	8′ 4½″	8′ 5¼″	8′ 5″	8′ 3¾″	8′ 4⅝″	8′ 5⅜″
Displacement, tons	742	750	748	737	747	757
Trim by the stern, inches	2¼	⅝	1½	9¼	7¾	5⅜
Speed knots per hour	12·11	16·08	20·04	25·10	27·57	29·69
Mean revolutions	277	378	469	617	697	782
Mean slip, per cent.	7·59	10·80	10·26	14·58	17·15	20·52
Boiler pressure, lb.	244	251	250	248	—	244
Number of boilers in use	1	2	2	4	4	4
Shaft horse-power	562	1564	2848	6351	9333	12622

THE U.S. TORPEDO-BOAT DESTROYER "MONAGHAN."

By Commander U. T. HOLMES, U.S. Navy.

(*American Society of Naval Engineers; August*, 1911.)

THE contract for the *John R. Monaghan* was made with the Newport News Shipbuilding and Dry Dock Co. on the 23rd June, 1909. The keel was laid on the 1st June, 1910, and the vessel was launched on the 18th February, 1911. The official trials were run in May, 1911.

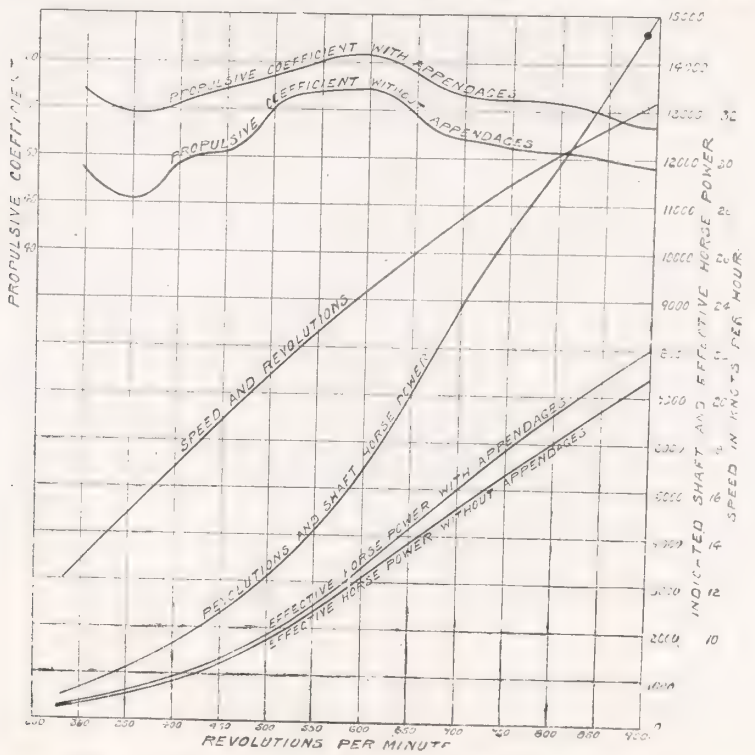

PROPULSIVE COEFFICIENT WITH APPENDAGES
PROPULSIVE COEFFICIENT WITHOUT APPENDAGES
SPEED AND REVOLUTIONS
REVOLUTIONS AND SHAFT HORSE POWER
EFFECTIVE HORSE POWER WITH APPENDAGES
EFFECTIVE HORSE POWER WITHOUT APPENDAGES
REVOLUTIONS PER MINUTE
INDICATED SHAFT AND EFFECTIVE HORSE POWER
SPEED IN KNOTS PER HOUR

Fig. 1.—Standardization Trials, U.S.S. "Monaghan."

The general particulars of the vessel are :—

Length B.P. on load waterline ...	289ft. 0in.
Breadth ...	26ft. 4½in.
Depth moulded ...	16ft. 1in.
Draught ...	8ft. 4in.
Displacement ...	742 tons.
Coefficient of fineness, block ...	·41
Midship section ...	·66
L.W.L. ...	·66
Cylindrical ...	·63

The machinery consists of five ahead turbines and two astern turbines of the usual Parsons type, arranged on three shafts. The boiler installation consists of four water-tube oil-burning boilers of the Thornycroft type, arranged in two boiler rooms. The results of the official standardization trials are shown in Fig. 1, and the main results of the various 4-hour trials are given in Table I.

TABLE I.—OFFICIAL 4-HOUR TRIALS, N.S.S. "MONAGHAN."

Mean Draughts ...	8′ 5½″	8′ 4½″	8′ 4⅝″	8′ 4⅝″	8′ 3⅞″	8′ 4″
Displacement, tons ...	748	740	736	736	733	735
Trim by the stern, inches ...	11¼	11¾	15	14	13	16½
Speed knots per hour ...	11·93	16·12	20·12	25·32	27·76	30·45
Mean revolutions ...	282	380	480	634	709	817
Mean slip, per cent. ...	11·05	10·75	11·14	15·67	17·95	21·73
Boiler pressure, lb. ...	237	240	258	259	253	235
Number of boilers in use ...	1	1	2	4	4	4
Shaft H.P. ...	512	1337	2700	6628	9239	12410
Water consumption per hour per S.H.P. of main shafts, all machinery, lb. ...	43·06	24·00	17·99	15·07	16·20	15·49
Fuel oil consumption per S.H.P. per hour, lb. ...	3·79	2·21	1·61	1·33	1·33	1·33

Trials of Merchant Ships.

THE GEARED TURBINE CHANNEL STEAMERS "NORMANNIA" AND "HANTONIA."

By Professor J. H. Biles, LL.D., D.Sc.

(*Institution of Naval Architects; March*, 1912.)

THESE vessels have been built by the Fairfield Shipbuilding and Engineering Co. to the order of the London and South Western Railway Company. The following are the leading dimensions :—

Length overall	299ft. 0in
Length B.P.	290ft. 0in.
Breadth moulded	36ft. 0in.
Depth moulded to promenade deck	23ft. 6in.
Boilers	1 D.E. & 1 S.E.
Length of double-ended	21ft. 10in.
Length of single-ended	11ft. 4in.
Diameter	17ft. 0in.
Total grate surface	303 sq. ft.
Total heating surface	10,221 sq. ft.
Pressure	160lb.

The propelling machinery consists of four small quick-running turbines driving twin propellers by means of mechanical reduction gearing. One propelling unit is shown in Fig. 1, and consists of a high-pressure turbine running at about 2,000 revolutions per minute, and a combined low-pressure and astern turbine running at about 1,500 revolutions per minute, which drive the propeller shaft at about 315 r.p.m. by means of the small and large pinion wheels shown in the illustration. The diameter of each propeller is 8ft. The object of the installation is to secure greater economy than would be derived from a direct-driven turbine arrangement.

The trials of the *Normannia* took place in the Firth of Clyde, the relation between revolutions

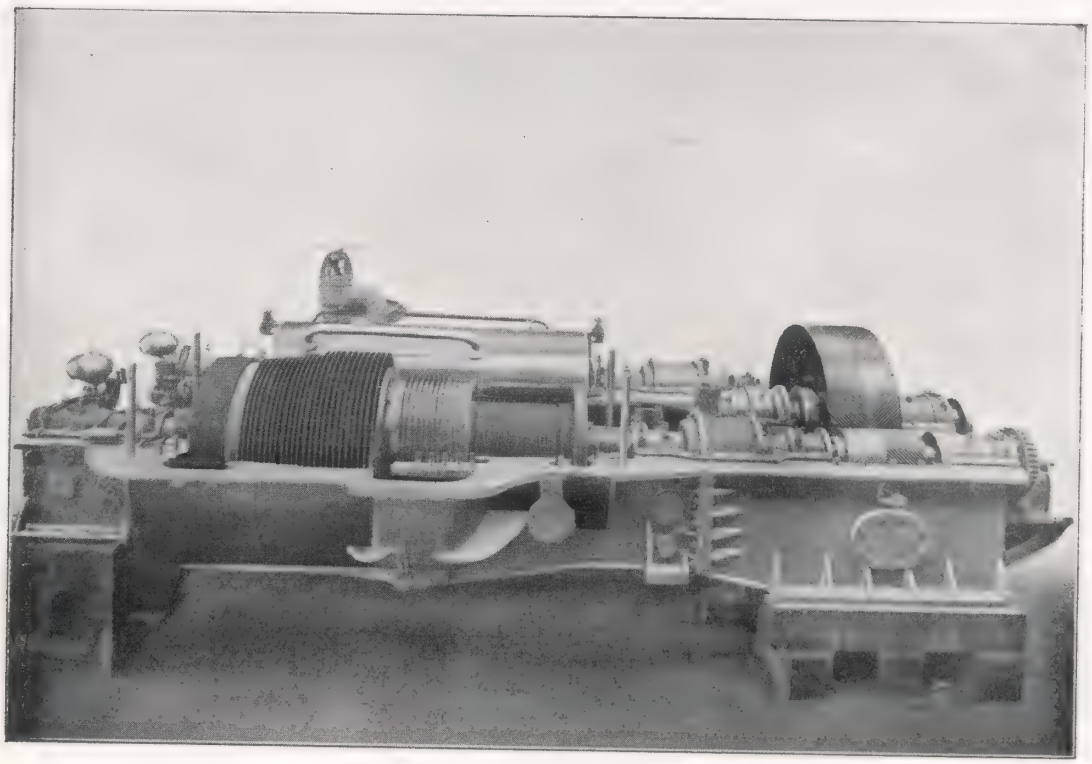


Fig. 1.—Geared Turbines of S.S. "Normannia" and "Hantonia."

and speed being first established by progressive runs on the measured mile. The maximum speed at which the vessel was run was 20·4 knots, the corresponding shaft horse-power being 6,100. To do the contract speed of 19½ knots, it was shown that 4,750 S.H.P. would be required. The vessel was run for six hours, and maintained an average of 5,000 S.H.P., which corresponds to a speed of 19·7 knots. The average air pressure in the stokeholds was less than ½-in.

The total amount of coal burnt during the six hours' trial was carefully weighed out, and showed a consumption per S.H.P.-hour of 1·34 lb. The water consumption for all purposes was 14·3 lb. per S.H.P.-hour. As the water consumption of the auxiliaries was 2·3 lb. per S.H.P.-hour, that for the main engines was only 14·3 – 2·3 = 12 lb. per S.H.P.-hour. In the case of two direct-driven turbine ships, the *Cæsarea* and *Sarnia*, completed for the same owners in 1910–11, the water consumption for the main engines was 15·1 lb. per S.H.P.-hour, so that in the later vessels a notable economy has been achieved in this respect.

The trials of the *Normannia* and her passage from the Clyde to Southampton have proved that no unpleasant noise is caused by the gearing. By listening carefully when in the passenger quarters a slight whistling sound, something like a small steam escape, can be heard. Professor Biles also states that there is absolute freedom from vibration.

PERFORMANCE ON SERVICE OF THE CHANNEL STEAMER "NEWHAVEN."

By Monsieur P. Sigaudy.

(*Institution of Naval Architects; March*, 1912.)

THIS paper contains the service results of the steamer *Newhaven* during the first five months of her running on the route between Newhaven and Dieppe. The dimensions of the vessel are :—

Length overall	302ft. 0in.
Length B.P.	292ft. 0in.
Breadth	34ft. 7in.
Depth	22ft. 2in.
Maximum draught	9ft. 8in.

Passenger accommodation for 1,000 persons.

The machinery consists of three Parsons turbines, one h.p. turbine driving the centre screw, and two l.p. turbines driving the two wing screws. The reversing is done on the two wing screws. The boilers are of the water-tube type. The starting platform is on the main deck level in a roomy and well-lighted position, in which all the controls are assembled. The main regulating valve, the two manœuvring valves, the by-pass, the heating-through valves, and the lubricating system are all handled from this central position. The change-over from ahead to astern, or the contrary, is effected in four seconds, and the vessel can be stopped in a distance of 295 feet from a speed ahead of 12 knots.

The contract trials consisted of a double crossing between Dieppe and Newhaven, at a time of tide selected by the owners in order to neutralize the effect of currents. The duration of the run from Dieppe to Newhaven was 2 hours 43 min. 51 sec., and from Newhaven to Dieppe 2 hours 43 min. 5 sec., giving a mean speed of 23·85 knots, the distance from light on pierhead to light on pierhead being 65 nautical miles. These trials were carried out on the 26th and 27th May, 1911.

The *Newhaven* was put on service on the 2nd June, and ran till the 20th October inclusive, being then withdrawn for inspection at the expiration of the guarantee period. Between the above dates she made 126 crossings, the mean time of which was 2 hours 51 min. 25 sec., corresponding to a mean speed of 22·70 knots. The fastest trip occupied 2 hours 43 min. (slightly less than the trial runs), with a favourable tide. The slowest trip took 3 hours 6 min., in bad weather. The mean coal per trip was 29 tons 4 cwt., including that consumed in raising steam and maintaining the fires in harbour. Although the author was not able to obtain exact information, he estimates that this quantity of coal may be divided as follows :—5 tons 1½ cwt. for raising steam and losses on arrival; 24 tons 2½ cwt. for the trip from pierhead to pierhead.

These results are notably better than those anticipated by the owners, who stipulated that the crossings should be made in 3 hours on the official trials. The author attributes them, in the first place, to the turbines of the Parsons type, and, secondly, to the special arrangements adopted for the auxiliary machinery, which was designed on a destroyer basis, with additional scantlings to ensure good wearing qualities.

The Strength and Elasticity of Materials.

THE STRENGTH OF MATERIALS.

By Professor W. C. Unwin, LL.D., F.R.S.

(*Presidential Address to the Institution of Civil Engineers; November*, 1911.)

In the course of his presidential address Professor Unwin dealt mainly with the subject of the strength of materials. He said that the object of a study of the strength of materials is to determine the proper dimensions to be given to parts of machines or structures, in order that they may resist the straining actions to which they are subjected without breaking or prejudicial deformation. The oldest and still the most common method of proceeding is to reduce the straining actions to simple tension, thrust, or shear, by calculations based on the assumption of Hooke's law, and to provide material enough to limit the intensity of these stresses to a fraction of the breaking strength of test-pieces similarly strained. The ratio of the breaking strength to the working strength is termed the factor of safety.

It is easy to show that the ratio of the breaking stress to the working stress is not a real factor of safety. The point at which danger occurs in different cases is not a fixed fraction of either the real or the calculated breaking load. The factor must be varied for different conditions, and can only be fixed empirically.

The ductile methods chiefly used in construction yield or suffer a large deformation at about half to two-thirds of the breaking stress. If the yield stresses are taken as the limits of safety, then the real factor of safety is only about half the ratio of the breaking to the working stress. Should, then, the yield point be adopted instead of the breaking strength as the measure of the constructive value of the material? With regard to this question, Professor Unwin drew attention to the fact that many constructive materials, such as stone, timber, and cast iron, have no yield point. Even for ductile rolled materials, such as mild steel, the point ordinarily determined as the yield point is not a very fixed point for a given material. It depends a little on the rate of loading. Yielding really begins at the elastic limit, a point below the yield point, spreading along the bar and becoming general over the bar at the yield point.

In a simple uniform tie-bar, no doubt, the deformation at the yield point fixes a limit to the possible working stress, but in most other cases only indefinitely short portions of a structure are exposed to the maximum stress, and the deformation due to these is not necessarily serious. In a riveted girder, for instance, designed for a limiting stress on the planes of weakness through the rivet holes, only very short portions of material near the planes of weakness reach the maximum stress under an increasing load, and the great mass of the material is subjected to much lower stresses. In fact, no marked yield point is shown in testing a riveted girder. In a riveted joint, the effect of slight yielding of short portions near the rivet holes is to render the distribution of stress more uniform. The yield point of the material does not appear to the author to measure a definite limit of safe load in such cases.

Some writers assume that the elastic limit, as determined in a tension test, is the real measure of the constructive strength of a material; but the author is not sure that this is not, when rigidly examined, the most ambiguous and elusive of all the measures proposed. Cast iron has no elastic limit, nor have copper and most alloys, unless hardened by straining, cold rolling, or drawing. In mild steel there is a primitive tensile elastic limit, or limit of proportionality, determinable by careful measurement, but it is not an absolutely fixed point. Its position can be altered by straining, and its primitive position is, in general, a somewhat artificially raised one due to the process of manufacture.

No doubt there are cases where the primitive tensile elastic limit does fix a superior limit to the stress consistent with safety, but it cannot be taken generally as an exact measure of constructive value, although it certainly is a valuable indication. Further, the elastic limit in compression is seldom observed, and the elastic limit in shear is experimentally almost unknown; yet resistance to shear is probably for ductile material the most important element of constructional value.

The classical research of Wöhler showed that,

Photo by] Professor W. CAWTHORNE UNWIN, LL.D., F.R.S., [*Fred. Hollyer, Kensington.*
President of the Institution of Civil Engineers.

W. Cawthorne Unwin

subjected to equal alternate tensions and thrusts, a steel bar would in time fracture if the stress exceeded about one-third of the statical breaking strength. The ratio is, however, not a constant one, and must be determined independently for different materials or different qualities of the same material. Bauschinger showed that a bar, subject to equal alternating stresses gradually increased, would be found to have ultimately equal elastic limits in tension and thrust of about one-third the statical breaking weight, and these he termed " natural elastic limits." These limits do appear to define the range of variation of stress a material can sustain, but much is not known about them except in the case of steel. It is when these limits are exceeded in alternating stress that fatigue ensues and the bar breaks. Dr. Stanton's results confirm this. For the engineer's purpose, however, the natural elastic limits are not easily found.

Hence, whether the breaking stress, the yield stress, or the primitive elastic limit be taken, a satisfactory rational basis for measuring the constructive strength of a material has not been found. There is not a fixed ratio between the greatest safe working stress and any of these. Valuable as are the data of breaking strength, yield point, and elastic limit determined in a tension test in influencing the judgment of the engineer, they do not furnish any purely rational rule for fixing working stress for designers. Experience in similar cases must always be the ultimate appeal.

Another of the unsolved problems of engineering is the criterion of safety in cases of compound stress in which tension, thrust, and shear are combined. In such cases the theory of elasticity furnishes the principal stresses and the principal strains, but there is a difference of opinion among theorists as to which of these determine fracture. For simple stresses either assumption leads to the same result, but it is not so with compound stress. Very early Conlomb suggested that fractures would be determined by shear, as is the case with cast iron under compression. The subject is now under investigation experimentally, and, speaking generally, it seems that for ductile materials the material gives way at some limit of shearing stress, while for brittle material in tension the principal stress is the best criterion of safety.

It is only when comparing two different materials that an engineer needs to consider relative constructive value, and then he bases his judgment on all the properties of the material. Ordinarily the safe working stress is fixed by experience in similar cases, theory helping mainly in explaining why, in different cases, different stresses are suitable. The history of engineering design is a history of experiments in construction, in which by trial and error right proportions have been found ; and in the settlement, the different relative importance of cost and weight enter into the question as well as considerations of mechanical strength. The fact that scientific data and rational or semi-rational formulæ are conveniently used should not be allowed to conceal the fact that an empirical element always enters into the solution.

New designing is really a process of comparison in which the engineer extends experience in known cases to new conditions, and the problem is, in what proportion must dimensions be varied to allow for differences of size, of material, of loading, of speed, and of form? Most practical problems in designing for strength reduce themselves, if traced to their foundations, to applications of the law of similarity. Generally two geometrically similar structures are equally safe with loads of the same intensity applied at corresponding places. Two pipes are equally stressed by a given fluid pressure if the diameters and thicknesses are in the same ratio. Two uniform flooring slabs similarly supported will carry equally well a given load per square foot if the lengths, widths, and thicknesses are in the same proportion. If all the linear dimensions of two girder bridges are in the ratio N, they will be equally stressed by total loads in the ratio N^2. So-called mechanical instinct is, no doubt, really reasoning based on the law of similarity.

It should be noted that the total load on a structure is made up of the useful load and the weight of the structure, and that the latter increases as the cube of the linear dimensions. Hence, as the strength of the structure, or total load it will carry, increases only as the square of the linear dimensions, the weight of useless load becomes a larger fraction, and the useful load a smaller fraction, as the size increases. A limit of size is reached at which the structure will only carry itself. Hence proportions will in general vary greatly with size. In the Britannia Bridge the depth of girder was about one-sixteenth of the span. To make the Forth Bridge possible, the depth of the cantilevers was about one-fifth of the span. In that bridge the possible limit of span in a steel structure was nearly reached.

In the case of machines another consideration enters. The weight stresses are not usually important, but the inertia stresses due to variation of the velocity or direction of motion are very important. The stresses in a fly-wheel rim due to radial acceleration are proportional to the square of the rim velocity, whatever the cross section. Hence fly-wheels are equally safe in

this respect if the rim velocity is the same or if the rotations per minute are inversely as the diameters.

The exceedingly interesting application of a law of similarity in determining the resistance of ships by model experiments is outside the questions treated in this address. It is by an application of the law of similarity that the Engineering Standards Committee have been able to define the forms of test-pieces, so as to give comparable results as to percentage of elongation.

EXPERIMENTS ON THE STRENGTH AND FATIGUE PROPERTIES OF WELDED JOINTS IN IRON AND STEEL.

By Mr. T. E. Stanton, D.Sc., and Mr. J. R. Pannell.

(*Institution of Civil Engineers; December*, 1911.)

THESE experiments were undertaken at the suggestion of Sir John Wolfe Barry in 1908, the object being to obtain a comparison of certain properties of welded joints—made by different processes and by different makers—with the corresponding properties of the unwelded material from which the joints were made. It was hoped that by this means some estimate of the general reliability and efficiency of modern welding processes might be arrived at. In order to obtain material for the work a circular letter was sent from the National Physical Laboratory to various engineers, inviting them to submit specimens of welded joints for testing. Sixteen firms responded to this invitation, and the total number of welded joints received was 167. The method of welding and treatment of the joints was left entirely to the makers, the only condition being that all specimens should be made from bars $1\frac{1}{4}$in. in diameter. The tests to which the joints were subjected were: first, a tensile test, and secondly, a fatigue test by the Wöhler method. The former included determinations of the elastic limit, the yield point, the maximum stress, the total elongation, and the general and local elongations.

For the rapid carrying out of the fatigue tests a machine was constructed in which the specimen was rotated at a speed of 2,200 reversals per minute. Owing to the results of a previous investigater with another type of machine having shown a decided reduction of fatigue strength with increased rapidity of stress-alternation, preliminary experiments were carried out to determine the amount of this effect for the particular type of machine used. The results showed that for speeds of 200 and 2,200 reversals per minute there was practically no difference in the fatigue strength. The assumption was therefore made that the apparent reduction of fatigue strength with rapidity of alternation in the case referred to was a characteristic of the machine used.

The mean results of the tensile tests on the welded joints, expressed as a percentage of the strength of the original material from which the joints were made, were :—

Hand-welded iron	89·3 per cent.
Hand-welded steel	81·6 „
Electrically-welded iron	89·2 „
Electrically-welded steel	93·4 „

Joints made by the oxy-acetylene process were also submitted by two makers, but the results were not comparable with those obtained by the hand or electric processes.

The various determinations made in the tensile tests showed a distinct want of uniformity in the material in the region of a weld, but the results of the fatigue tests proved that this does not materially affect its resistance to reversals of stress. When failure under alternating stresses of low value takes place, it is invariably due to a defect in the actual weld itself. The number of defective joints which were discovered in the whole investigation, however, leads to the broad conclusion that in important work, where the failure of any particular welded joint may involve serious damage to the structure, the subjection of each joint to a proof-load is still desirable.

THE ENDURANCE OF METALS.

Experiments on Rotating Beams at University College, London.

By Mr. E. M. Eden, Mr. W. N. Rose, and Mr. F. L. Cunningham.

(*Institution of Mechanical Engineers; October*, 1911.)

When a piece of metal is subjected to alternating or intermittent stress, the metal may fracture after a large number of alternations or repetitions of stress, although the maximum stress applied is considerably less than would be required to fracture the material with a single application.

The tests described in this paper were undertaken with a view to finding, if possible, the form of the stress-repetitions curve for any one material, and to determine the effect of speed or rate of alternation of stress on the stress-repetition curve and on the limiting resistance for an indefinite number of repetitions. The testing machine was specially designed and built with these objects in view, but the tests deal also with the comparative endurance of various materials and with the effect of rests, vibration, surface and form of test-piece, and annealing, on endurance. The results obtained for different materials and qualities of material are given in

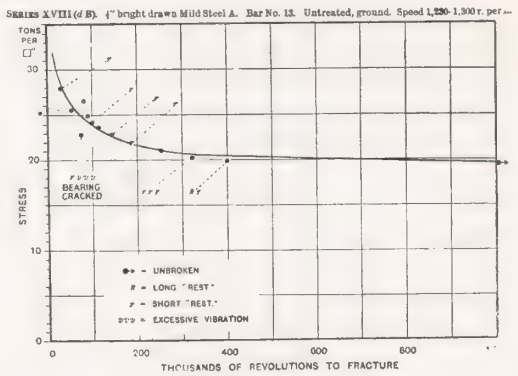


Fig. 1.

Table I.—Range of Stress.

Material.	Tensile Tests.		Endurance Tests.			
	Tenacity. Tons per Square Inch.	Limit of Elasticity. Tons per Square Inch.	Speed, Revolutions, or Alternations of Stress per Minute.	Range of Stress Causing Fracture after 10^6 Revolutions.	Ratio: Range / Tenacity.	Ratio: Range / Limit.
$\frac{7}{16}$-in. bright-drawn mild-steel rod, quality B	...	...	600 1,300	31·6	—	—
½-in. bright-drawn wrought-iron rod ...	33·8	26·5	250 620 1,300	34·6	1·02	1·3
½-in. bright-drawn mild-steel rod, quality A	35·7	25	300 600 1,300	39	1·09	1·56
Vanadium steel, annealed	38·4	26·8	600 1,300	41	1·07	1·53
25 per cent. nickel steel	48·8	24·1	1,300	44·2	0·90	1·83
Vanadium steel, oil-tempered	57·2	46·5	1,300	49	0·86	1·05
Copper trolley-wire, hard-drawn	...	...	1,300	22	—	—

a series of diagrams, one of which is reproduced in Fig. 1.

The range of stress in these tests is double the applied stress, as the stress alternates between equal positive and negative values. Table I. gives the values of this range of stress for some of the more important series of tests, and also the limit of elasticity and tenacity obtained from tensile tests of the materials. The endurance of the various materials, as shown by the range of stress figures, appears to depend more on the tenacity than on the limit of elasticity, the material with higher tenacity always showing greater resistance to alternating stresses. A comparison of these tests with alternating stress tests made by other experimenters is given in Table II. The most remarkable point shown in the University College tests is the absence of all speed effect between 250 and 1,300 alternations per minute.

TABLE II.—ENDURANCE UNDER ALTERNATING STRESSES.

Experimenter.	Type of Endurance Test.	Material.	Tensile Test Figures.		Endurance Test Figures.			
			Tenacity.	Limit of Elasticity.	Speed. Alternations of Stress per Minute.	Range of Stress causing Fracture after 10^6 Alternations.	Ratio: Range of Stress / Tenacity.	Ratio: Range of Stress. / Limit of Elasticity.
			tons per sq. in.	tons per sq. in.		tons per sq. in.		
Wöhler	Rotating cantilever	Phœnix iron	21·3	...	60—80	22·8	1·07	—
		Homogeneous iron ...	28·1	...	60—80	24·5	0·87	—
		Vickers' steel axles ...	27·7	...	60—80	24·0	0·87	—
		Firth's tool steel ...	55·0	...	60—80	30·9	0·56	—
Baker	Ditto	Soft steel	27·7	...	50—60	25·0	0·90	—
		Fine drift steel	54·0	...	50—60	27·5	0 51	—
Rogers	Ditto	Steel C (0·32 per cent C.) as rolled	29·3	16·7	400	32·0	1·09	1·92
		Steel C annealed ...	26·5	7·1	400	27·7	1·05	3·9
U.C.L.	Rotating beam uniform bending-moment.	½-in. bright-drawn wrought-iron bar	33·8	26·5	250	34·6	1·02	1·3
					620			
					1,300			
		½-in. bright-drawn mild-steel rod A	35·7	25·0	300	39·0	1·09	1·56
					600			
					1,300			
Turner	Fixed cantilever Rotating deflection	Mild steel	27·3	18·8	250	35·6	1·3	1·9
		Nickel steel	48·0	36·1	250	52·5	1·1	1·46
Stanton and Bairstow	Reciprocating weight, direct tension and compression. Ratio: Tension / Compression = 1·4	Wrought-iron No. 2 piston-rod steel	25·6	13·4	800	19·2	0·75	1·43
			43·8	19·6	800	28·3	0·65	1·44
Reynolds and Smith	Reciprocating weight, direct tension and compression. Ratio: Tension / Compression = 1·15	Mild steel annealed	25·8	...	1,337	20·9	0·81	—
					1,428	20·1	0·78	—
					1,516	19·2	0·75	—
					1,656	18·1	0·70	—
					1,744	15·2	0·59	—
					1,917	12·4	0·48	—
		Cast steel annealed	48·0	...	1,320	20·1	0 42	—
					1,660	18·3	0·38	—
					1,820	16·8	0·35	—
					1,990	13·1	0·27	—

LOAD-EXTENSION DIAGRAMS OBTAINED PHOTOGRAPHICALLY WITH AN AUTOMATIC SELF-CONTAINED LOAD-EXTENSION INDICATOR.

By Professor W. E. Dalby, M.A., B.Sc.

(*Institution of Naval Architects; March*, 1912.)

THE load-extension indicator illustrated in Fig. 1 is an instrument recently designed by the author, in which the effects of inertia and pencil friction are eliminated. When used to obtain a diagram, the instrument is detached from the tripod frame, the nut *A* of the frame being replaced by the upper shackle of the testing machine. The upper end of the specimen to be tested is screwed into the coupling *B*, and the lower end is screwed into the lower shackle of the testing machine. The specimen and the "weigh-bar" *C* of the instrument are thus placed in series, and have their cross sectional area so related that the breaking load for the specimen produces a stress on the weigh-bar well within the elastic limit. The extension of the weigh-bar is measured by the angular displacement of a mirror connected to it. The extension of the specimen is measured by the angular displacement of a second mirror, mounted on the axis *D* and connected to the specimen by linkage attached by the arm *E*. The two mirrors are placed with their axes at right angles. A source of light is arranged in the head *F*, a ray from which is reflected from each mirror in succession to the focussing plane of the camera *G*.

Typical load-extension diagrams for various materials as obtained by this instrument are given in Fig. 2. The apparatus brings out in a most interesting way the peculiar relation between the load and the extension in the neighbourhood of the yield point for the iron and steel samples. In the case of the copper specimen the diagram shows that the material simply gave way and drew out without any discontinuation in the load-extension curve, thus showing that the irregularities in the yield point regions of the iron and steel diagrams are not due to any peculiarities of the instrument. Another point of interest in connection with these diagrams is that the load on the specimen at the instant of fracture can be read off without any ambiguity.

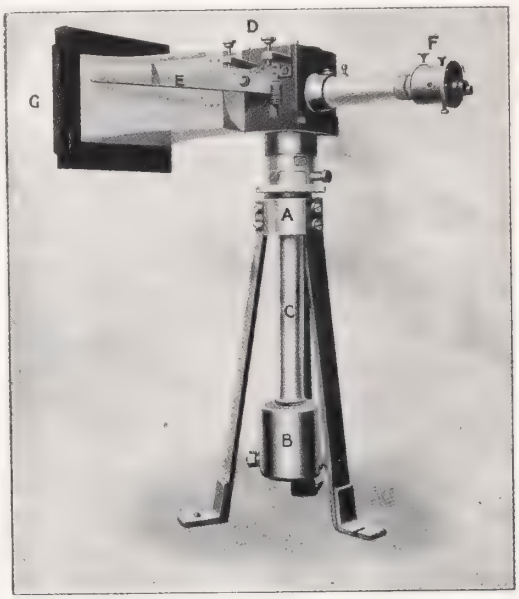


Fig. 1.—Optical Load-Extension Indicator.

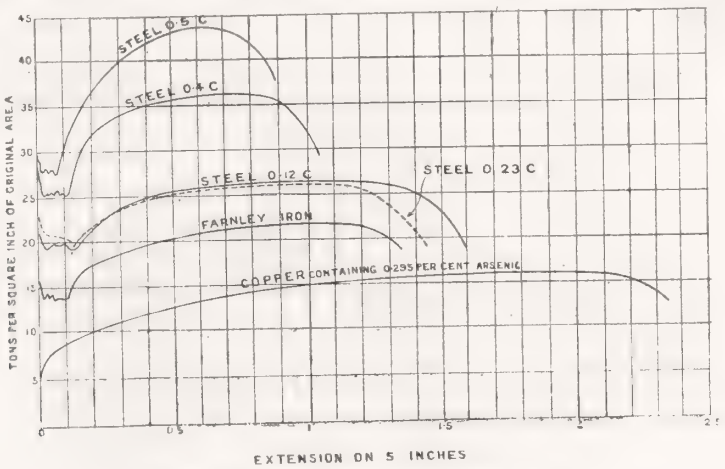


Fig. 2.—Load-Extension Diagrams.

THE STRENGTH OF STEEL TUBES, PIPES AND CYLINDERS UNDER INTERNAL FLUID PRESSURE.

By Mr. Reid T. Stewart.

(*American Society of Mechanical Engineers; April*, 1912.)

A COMPARISON of the theoretical formulæ for the strength of steel tubes, pipes, and cylinders under internal fluid pressure shows that :

a Clavarino's formula is theoretically correct for cylinders with attached heads, and Birnie's formula for heads held independently. These formulæ are practically applicable to certain classes of seamless steel tubes and cylinders and to critical examination of ordinary commercial steel tubes, pipes and cylinders when sufficiently accurate data are available.

b In commercial welded pipe the variation in thickness of wall, perfection of weld, etc., account for variation in bursting strength of sufficient magnitude to render unnecessary any consideration of Clavarino's or Birnie's condition of head support. All such variations give rise to errors on the side of danger when applying these formulæ.

c For ordinary commercial wrought pipe Barlow's formula is to be preferred. This formula is

$$\frac{p}{f}=2\frac{t}{D};\ p=2f\frac{t}{D};\ t=\tfrac{1}{2}D\frac{p}{f};\ f=\tfrac{1}{2}D\frac{p}{t}$$

where D = outside diameter in inches.
t = nominal or average thickness of wall in inches.
p = internal fluid pressure in lb. per square inch.
n = safety factor as based on ultimate strength.
f = fibre stress in lb. per square inch.
$= \frac{40{,}000}{n}$ for butt-welded steel pipe.
$= \frac{50{,}000}{n}$ for lap-welded steel pipe.
$= \frac{60{,}000}{n}$ for seamless steel tubes.
$= \frac{28{,}000}{n}$ for wrought-iron pipe.

THE USE OF POLARIZED LIGHT AS AN AID TO DETERMINING THE EFFECT OF STRESS IN ENGINEERING MATERIALS.

By Professor E. G. Coker, M.A., D.Sc.

(*Sheffield Society of Engineers and Metallurgists; December*, 1911.)

THE term polarization as applied to a wave of light is intended to convey that its transverse motion is executed in some particular plane. Thus, if a beam of ordinary light is reflected from the black glass mirror *AB* (Fig. 1), the reflected ray is polarized in a direction perpendicular to the plane of the paper, and has acquired directional properties which make it extremely useful for many purposes. The same result is obtained by allowing light to pass through certain crystals. Such polarized light affords a valuable experimental means of studying the stress distribution and intensity in certain materials under load.

If a beam of polarized light is passed through a plate of glass or other transparent material subjected to tension of a uniform amount, and then viewed by a suitable optical device, it will appear coloured, and these colours will change with each alteration of load. When a small load is applied to a tension member, made of a transparent material like celluloid, it causes the member to appear a faint white; and as the stress increases the colour of the specimen changes to a lemon yellow, and then through orange to a purple red. As the load is still increased it changes to a sky blue, and finally becomes white again. If the load is still further increased these colours occur again in the same order, and each is associated with a definite stress. All colours may be obtained at once if the specimen is bent, or otherwise subjected to varying stress.

The reason for the display in colour is due to the fact that a plane polarized beam of light *D*, such as is indicated in Fig. 2, as issuing from a Nicol's prism *P*, in general divides up into two components, *O* and *E*, within the stressed

material. These components have their planes of vibration at right angles to one another, and in the directions in which the stresses are respectively, a maximum and a minimum at any given point in the plate. Each wave also suffers a retardation in the material, but not of the same amount, and this causes a difference in phase which persists in their subsequent propagation through the air. A Nicol's prism, *A*, shown interposed in this beam, rejects all components

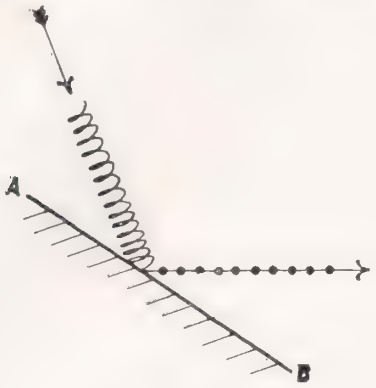


Fig. 1.

of the transverse wave motion except those in one particular plane, and the resulting waves which finally emerge give an interference effect which produces a colour impression to the eye.

Professor Coker then deals with the application of the method to some simple examples. He considers that results obtained with glass, and to a lesser extent celluloid, could be applied with confidence to infer the stress distribution if the material were iron or steel. In the case of a beam bent by a uniformly applied moment, a dark band is seen at the centre with parallel colour bands above and below, thus showing that the stress is proportional to the distance from the neutral axis—a well established result in the theory of beams. When another similar and similarly bent beam is placed alongside, the joint effect is seen to cause a doubling of the bands—a result which indicates that the optical effects of tension stresses and compression stresses in the same direction may be added. If the tension side of one beam is placed next the compression side of the other, a uniform dark field is obtained, showing that the interference effects practically neutralize one another, and it may be inferred, therefore, that the colour effect of a tension stress annuls the colour effect of a compression stress of the same magnitude if they are both in the same direction.

This is only true, however, for this particular direction, for if the beams are placed at right angles brilliant colour bands are obtained in the common area, showing that stresses in the same plane and at right angles to one another produce effects which require to be examined separately. In this case a black band traverses the common area in a nearly diagonal direction, passing everywhere through points where the stresses in the two beams are equal. It follows that a black band in the stressed material may denote no stress at all or equal stresses at right angles.

The experiments, therefore, show that the colour effects depend on the directions of the stresses as well as on their magnitudes. If the stresses are in the same direction, their optical effects are added ; and if they are at right angles, their optical effects are subtracted if due account is taken of the sign of each stress. This last result is specially important, since the stress at any point in a plane lamina can always be represented by two stresses at right angles, and the optical effect measures their difference, or what is in effect the shear at the point considered.

A subject of general interest in practical construction relates to the effect of holes, notches, and other discontinuities on the strength of the structure in which they occur. One of the simplest cases arises when a hole is drilled in the central line of a tension bar. When viewed by the author's apparatus it is seen that the stresses produced in the neighbourhood of the hole vary greatly, especially at the minimum cross section, rising to approximately three times the minimum stress at the point *A* (Fig. 3) when the hole is not very great compared with the breadth of the

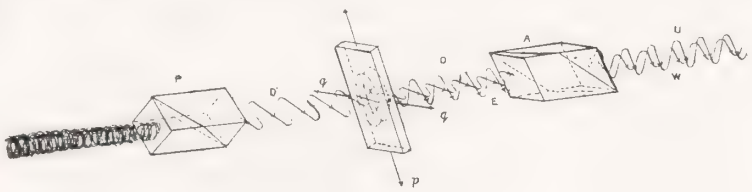


Fig. 2.

specimen. The maximum stress p_1 is given approximately by—

$$p_1 = \frac{6c^3}{2c^3 + 2c^2 + c + 1} p$$

where p is the average stress obtained by dividing the total load by the minimum cross sectional area and c is the ratio of the breadth of the member to the diameter of the hole. If c is large this expression reduces to

$$p_1 = \frac{3c}{c + 1} p.$$

It may, however, be doubted if this maximum stress is always reached in practice, as, owing to the great intensity of the stress in the neighbourhood of the periphery of the hole, some plastic deformation probably occurs and a re-distribution

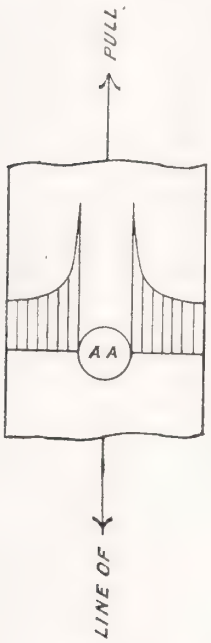


Fig. 3.

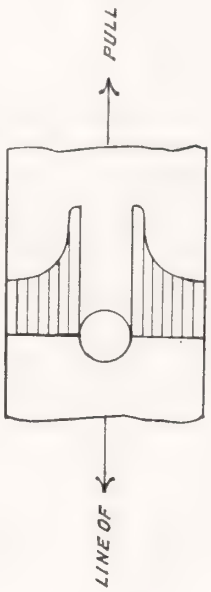


Fig. 4.

of the stress will then take place; so that a condition is obtained in which the stress is probably fairly uniform close to the hole, as indicated in Fig. 4, and then rapidly falls away in intensity, but is probably slightly greater than before owing to the plastic re-distribution.

Another example of a discontinuity in a tension member is that of two semi-circular notches arranged symmetrically as in Fig. 5. If the section is not reduced very much by the notches, it is found that the distribution of tension across the minimum section is such that the maximum stress is very nearly twice the minimum value (Fig. 5). The effects of notches of this kind, therefore, are not so serious as a hole in the plate, but the maximum stress may easily rise so much that plastic yielding takes place with a probable

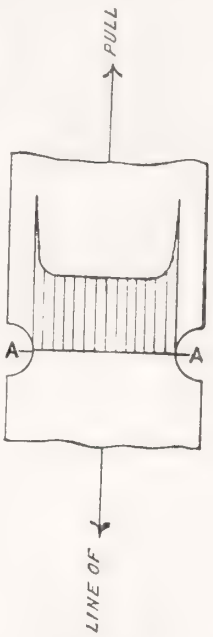


Fig. 5.

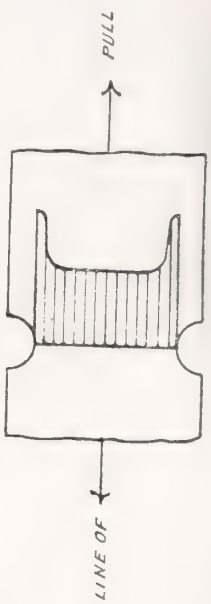


Fig. 6.

re-distribution of stress like that shown in Fig. 6. The latter result has a bearing on the mechanical testing of materials, as it shows clearly that a test bar which approximates to this form will begin to

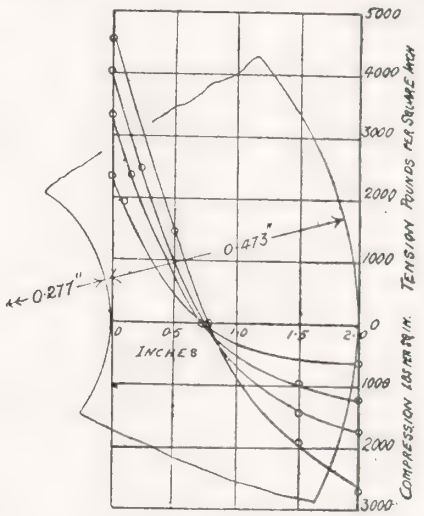


Fig. 7.

fail at the points A long before the average stress intensity reaches the yield point of the material in simple tension, and any deductions based upon

the assumption that the stress in a tension bar of variable section is uniform at the minimum section is likely to be in error by a large amount.

In actual construction the conditions of stress which arise are usually much more complicated. If, for example, a rivet fills the hole in a tension member the optical effects show that a considerable change is produced in the stress distribution. New lines of colour spring into view indicating increased stress at the minimum section, which from their shape and form lead to the inference that a large part of the increase is due to radial stress. In a riveted joint the stresses are still more complicated because the load is in general transmitted through the rivet. Such a distribution of stress is one of the simplest of those actually occurring in practical work, but, so far as the author is aware, it has not yet been accurately determined either experimentally or by analysis.

Another subject of interest, which is often of great importance in practical work, relates to the behaviour of curved members like hooks, links of chains, corrugated boiler flues, arched rings of bridges, and the like, in which the curvature of the member is an important factor in the distribution of stress due to an applied load. The stress distribution obtained by the optical method in the case of a curved beam (Fig. 7) shows that the usual formulæ which are used to calculate the stresses in straight beams are not applicable to curved ones, since the stresses no longer vary linearly across the section, and the position of the neutral axis is also not the same as in a straight beam, which is subjected to a load causing direct tension and compression.

Several colour photographs are attached to the paper showing the results obtained in typical cases.

THE DISTRIBUTION OF STRESS IN PLATES HAVING DISCONTINUITIES, AND SOME PROBLEMS CONNECTED THEREWITH.

By Dr. K. Suyehiro.

(*Japanese Institution of Naval Architects; November*, 1911.)

THE author has succeeded in solving mathematically the case of a tension strap pierced with a circular hole. The component stresses in such a plate, expressed in polar co-ordinates, were shown to be as follows :—

$$P = \frac{T}{2}\left(1 - \frac{a^2}{r^2}\right) + \frac{T}{2}\left(1 - 4\frac{a^2}{r^2} + 3\frac{a^4}{r^4}\right)\cos 2\theta$$

$$Q = \frac{T}{2}\left(1 + \frac{a^2}{r^2}\right) - \frac{T}{2}\left(1 + 3\frac{a^4}{r^4}\right)\cos 2\theta$$

$$R = -\frac{T}{2}\left(1 + 2\frac{a^2}{r^2} - 3\frac{a^4}{r^4}\right)\sin 2\theta$$

where P, Q, R are the radial normal stress, the transverse normal stress, and the shearing stress along the radial or transversal plane respectively; a is the radius of the pierced hole; and T the edge traction. From these equations it was proved that on the very periphery of the hole there occurs concentration of stress of an intensity three times the mean stress. The distribution of stresses is shown, for the weakest section, in Fig. 1.

The mathematical solution was verified by means of an india-rubber model. Fig. 1 shows the fairly close agreement between the mathematical and experimental results.

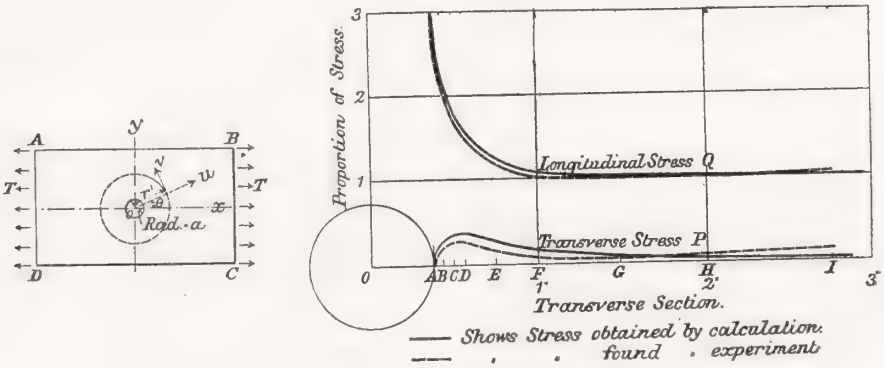


Fig. 1.

With regard to the breaking load, test bars of ductile metal, such as mild steel, pierced with a hole, are found to be stronger than those without the hole, if they are of ordinary length; but if the length is very short, the ultimate load is found to fall off. This may be explained by the hindrance to the free flow of metal.

Though the author has had no opportunity of testing pierced bars under repeated alternating loads of tension and compression, there can, he thinks, be no doubt that the lowering of ultimate load would, in this case, be very marked.

The longitudinal members of a ship subjected to severe stress may present features similar to the last two cases, because in them the stresses are alternating and repeated, and the flow of metal, which in a more simple structure occurs when the elastic limit is once exceeded, is hindered by the transverse members of the ship.

The similarity of stream lines and stress lines has very often been the subject of controversy among naval architects. The author disproves similarity in the two cases by the foregoing equations, and also points out that even the closeness of adjacent stress lines is not a direct indication of the stress intensity in a strained elastic plate.

AN ELECTRICAL TORSIONMETER.

By Herr Fritz Lux.

(*Schiffbautechnische Gesellschaft; November*, 1911.)

THE principle of the torsionmeter devised by the author is illustrated by Fig. 1. The shaft *1*, whose angle of torsion is to be measured between the points *2* and *3*, is provided with a pointer *4* and a stud *5*. At each revolution of the shaft the stud *5* comes in contact with the knife *6*, completing a primary circuit, in which

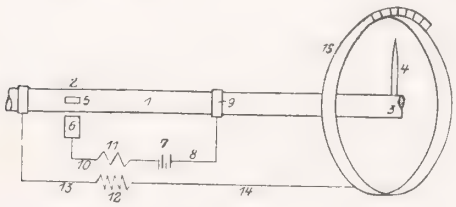


Fig. 1.—Electrical Torsionmeter.

the current flows from the battery *7* through the conductor *8*, slip-ring *9*, shaft *1*, stud *5*, knife *6*, conductor *10*, primary coil *11*, and back to the battery again. Once in each revolution the primary coil *11* induces a high-tension current in the secondary coil *12*, the secondary circuit being formed by the conductor *13*, shaft *1*, pointer *4*, graduated arc *15*, conductor *14*, and coil *12*. The current passes as a spark between the pointer *4* and the graduated arc *15*. When the shaft is not strained the spark always passes at the zero point on the graduated arc, but when twisting occurs the position of the spark is displaced from the zero, the amount of displacement giving the angle of torsion. By having several studs and pointers on the shaft the torsion can be measured at a corresponding number of points in each revolution, thus showing whether torsional vibrations exist or whether the torsion during one revolution is uniform. The instrument can be arranged to record the torsion upon a revolving drum, or, by introducing suitable mechanism, can be made to register the product of the number of revolutions and angle of torsion, giving the readings in H.P.-hours, if, when fixing the gear ratio of the mechanism, the modulus of rigidity is also taken into consideration. These recording devices are fully described in the paper.

THE CORROSION OF BRASS, WITH SPECIAL REFERENCE TO CONDENSER TUBES.

By Mr. P. T. Brühl, M.Sc.

(*Institute of Metals; September*, 1911.)

THE corrosion of brass condenser tubes may conveniently be classed under four heads:—

(1) *Corrosion proper.* Where the metal is uniformly removed over the whole surface.

(2) *Dezincification.* Where the zinc being

preferentially more rapidly removed than the copper, the surface of the metal becomes copper-rich in places.

(3) *Pitting.* This is due to galvanic action between the deposit on and the metal of the condenser tube, and is by far the most serious form of corrosion as it considerably curtails the life of the tube.

(4) *Erosion.* Here the mechanical influence of the stream of cooling water comes into play. Its effect is usually small and uniform.

Five selected cases of corroded condenser tubes are described by the author to illustrate the effects observed in practice. The examination of these tubes seemed to show that iron hydrate is by far the most injurious constituent of the deposit, being more electro-negative than cupric oxide and the basic chloride. A remedy which suggests itself immediately is the use of some protective coating, which should be electro-negative preferably to iron hydrate. Various protective coatings have been proposed, but the author is of opinion that on the whole the question of lessening corrosion by such coatings does not promise to be of economic usefulness in the case of condenser tubes. The line along which future investigation ought to be made is the study of the alloys of brass with one or even more other substances, with a view to obtaining a mixture which will give the requisite mechanical properties, and at the same time be the least influenced by sea water. The principal evils to be contended against are iron hydrate and carbonaceous matter. It is admitted that vessels which spend the greater portion of their time at sea beyond the limits where town sewage and river dirt contaminate the water experience fewer troubles from pitting than vessels which ply up and down foul rivers. It follows that the marine engineer should make a point of flushing out his condenser with clean water on arriving in port. The iron hydrate is derived no doubt from the cast iron end cover and the inlet pipe, although scale may be drawn in with the cooling water. It would appear advisable, therefore, for the end cover and inlet pipe to be made of brass. Further, the use of some form of filter for the incoming water would be advantageous.

Other matters affecting corrosion are then dealt with as follows :—

Effect of Strain. The irregular distribution of strain on any metallic surface gives rise to differences of potential. Some experiments made to investigate this point proved that, although a condenser tube does exhibit differences of potential along its length, these differences are not sufficiently large and abrupt to explain any corrosion other than a more or less uniform one extending over the whole surface.

Effect of Crystal Size. To test this matter two sets of plates of identical composition were so treated that in one set much smaller crystals were obtained than in the other. When immersed is sea water it was found that during the first six weeks the plates with the small crystalline structure showed the most corrosion, but that after some twelve weeks' immersion the small crystalline plates commenced to lose weight more slowly than the large crystalline plates. No explanation was found for this change. The method of manufacture of condenser tubes does not, however, permit of serious variations in the crystalline structure, and therefore crystal size plays a very insignificant part in the problem of corrosion.

Effect of Different Annealing Temperature. The author's experiments showed that corrosion increases with the temperature of annealing. It is not good, therefore, for the final annealing in the manufacture of condenser tubes to be unduly long.

Effect of Rolling. Brass, on rolling, becomes more electro-positive, or, in other words, more liable to corrosion, than the soft metal. Experiments with three plates of 70/30 brass, hard rolled to different extents and then immersed in sea water, showed clearly how the rate of corrosion increases with the amount of work expended in rolling. In view of the more rapid corrosion of hard-rolled metal, it would appear advisable, considering the thinness of the tube wall, that the last pinch in the drawing of condenser tubes (after the final annealing has been performed) should be as small as possible.

Effect of Varying Conditions met with in Practice. A series of experiments were undertaken to test the rate of corrosion under varying conditions, such as are met with in practice. The results leave no doubt that a rise of temperature up to a certain extent is favourable to corrosion, as it increases the reaction velocity between the brass and the sea water. An increase in the quantity of dissolved air also promotes chemical activity.

Effect of Oil. The oil used in the lubrication of the piston and slide valves should be composed of hydrocarbons only, that is to say, it ought always to be of mineral origin. Organic oils are subject to partial decomposition in the presence of steam, giving rise to glycerine and free acid, which latter becomes a serious danger when in contact with the wall of the condenser tube.

Effect of Stray Electric Currents. There can be no doubt that stray currents of electricity

escaping from imperfect fittings increase the rate of corrosion of condenser tubes by increasing the velocity of removal from the solution of the charged ions. The prevention of corrosion from this source can be effected by providing a good path for the stray currents outside the condenser.

The Localities of Pits. The positions of the more serious pits are probably governed by two factors—the presence of mechanical defects such as "spills" and of certain deposits. Even a comparatively small "spill" in the tube wall will result, when the cooling water ceases to be pumped through, in the formation of a little reservoir, to which air has complete access. It is precisely under these conditions that corrosion becomes very marked. The second cause of pitting is the strong electro-chemical action set up by certain substances. Of these, ferric hydrate and carbon are probably the worst, and of the corrosion products of brass the oxy-chloride appears to be the most injurious.

The Effect of Impurities in Brass. It was found that iron, nickel, and small amounts of lead are injurious, while tin up to about one per cent., large amounts of lead, and aluminium are useful in diminishing corrosion.

In conclusion, the author urges that so important a subject as corrosion of brass by sea water, the neglect of which must mean no slight expenditure, should induce steamship companies to keep records bearing on the subject.

THE CORROSION OF CONDENSER TUBES BY CONTACT WITH ELECTRO-NEGATIVE SUBSTANCES.

By Mr. Arnold Philip, B.Sc.
(*Institute of Metals; January*, 1912.)

ALL condenser tubes through or round which sea water flows are corroded, but the nature of this corrosion varies very greatly. By far the greater number of such tubes are merely uniformly corroded over as much of their surface as is exposed to salt water. A certain small number of tubes also undergo a local corrosion or pitting. The author's remarks apply specially to the relatively small number of cases in which localized corrosion occurs in condenser tubes made of Admiralty composition, *viz.*, copper 70, tin 1, and zinc 29 parts per cent. In the author's opinion the main problems which remain to be solved concerning localized condenser-tube corrosion are, firstly, the explanation of the cause of rather less than 10 per cent. of the small number of cases which are now observed; and, secondly, the device of a general means of preventing these and all other cases of corrosion superior to the method of protector bars, etc., of zinc, aluminium, steel, or iron, as at present employed. The presence of a so-called electro-positive protective metal such as aluminium, zinc, or iron, is not, however, alone sufficient to prevent the corrosion of the condenser tubes immersed in the same bath of sea water. It is essential that there shall be a direct electrically conducting connection between the tubes and the protecting metal or metals, and that the protector metal and the condenser tubes shall be both immersed in the same electrolyte.

The ordinary normal length of life for the main condenser tubes of a battleship should be from ten to twelve years, and many such cases can be cited. There are, however, cases where it has been found necessary to renew the whole or part of the tubes after two years' service or even less. A pecularity of the small number of cases in which corroded condenser tubes have been observed is the fact that in almost all the tubes the corrosion has occurred along the bottom of the inside of the tubes, and it is considered that this indicates the trouble has been set up by the contact of particles of coke, carbon, ferric oxide, or other conducting electro-negative solid materials, deposited from the cooling water upon the inside bottom surfaces of the tubes, setting up local galvanic action. Such action is, of course, increased when a condenser is not in use and the tubes remain full of sea water, for the particles then have time to settle out along the bottom line and give rise to the localized corrosion. If, on the other hand, the condenser is emptied of sea water when out of use, there would again appear to be a possible source of danger of corrosion along the bottom inside surface of the tubes, owing to the fact that this portion would be longest exposed to the joint action of salt water and air. In either method of treatment the corrosion could be minimized or altogether removed if it were found feasible to employ condensers with vertical or strongly inclined tubes.

Five cases of corrosion of condenser tubes are all that have come under the author's own observation during the past three years, and in only one of these it is not possible to ascribe the corrosion to the contact of particles of electro-negative solid conducting material.

Miscellaneous Papers.

WIRELESS TELEGRAPHY, WITH PARTICULAR REFERENCE TO SHIP INSTALLATIONS.

By Herr Direktor Bredow.

(*Schiffbautechnische Gesellschaft; November*, 1911.)

THE paper contains an exhaustive description of the "Telefunken" system of wireless telegraphy. Numerous illustrations are given of the installations suitable for different classes of ships. The 3–4 K.W. type of station fitted on large passenger vessels is illustrated in Fig. 1. During 1911 the Telefunken Co. installed 122 stations of this pattern. The apparatus has a guaranteed transmission distance of at least 650 k.m. (400 miles) by day and 1,200 k.m. (750 miles) by night. The space occupied does not exceed 2 metres (6ft. 6in.) in length, width and height. The weight of the instruments is about 300 k.g. (660 lb.), and of the motor generator about 600 k.g. (1,320 lb.).

The motor generator, which for purposes of illustration is shown under the table, is usually placed on the deck outside the cabin. It consists of a direct-current motor (*S*) of 4–6 H.P. driving an alternator of 220 volts at a normal rate of revolutions of 1,500. The periodicity of the alternator is 500 per second, but for purposes

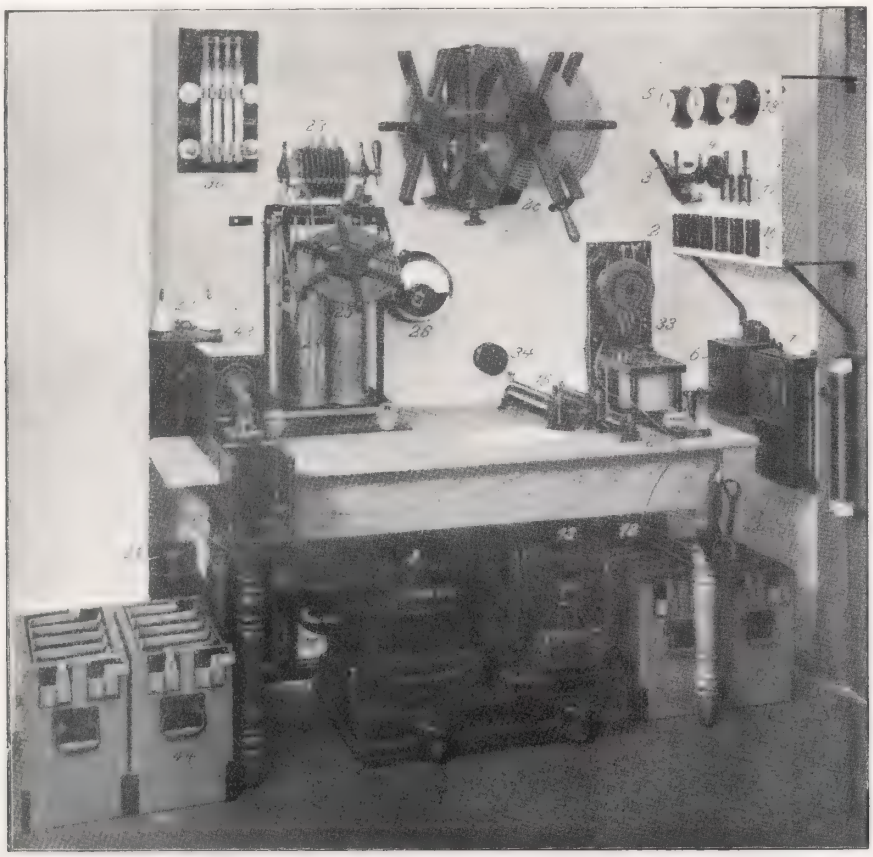


Fig. 1.—"Telefunken" Wireless Telegraphy Station for large Passenger Vessels.

of tuning the frequency can be varied through wide limits. The machines are protected against the high frequency of the transmitter by means of Leyden jars (*11*), also placed beneath the table. The generator is regulated by a sliding rheostat (*15*) placed on the table, by means of which the transmitter is tuned to the correct wave length. The various switches, safety fuses, and instruments required to control the motor and the distribution of the alternating current are arranged on a switchboard (*2–5* and *16–18*).

The alternating current at normal voltage delivered by the generator is transformed to a

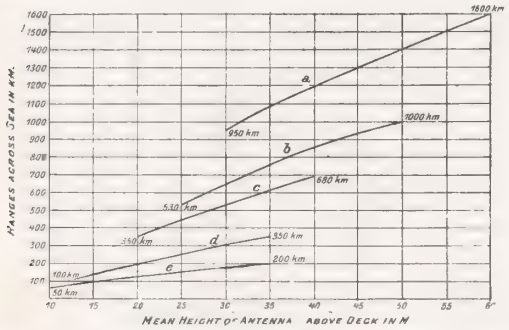


Fig. 2.—Transmission Distance, "Telefunken" System.

voltage of about 8,000 by the enclosed transformer (*22*). The choking coils (*27*) for tuning the low-frequency circuit to the high-frequency circuit are arranged beneath the table. The manipulating key (*20*) breaks the primary lead to the transformer. The transmitting apparatus is arranged in a simple and accessible manner. The multiple spark-gap (*23*) stands on an angle frame in which are placed the four Leyden jars (*24*) included in the primary circuit of the transmitter. In the front are placed the self-inductance coil of this circuit (*25*) and the plug connections for obtaining the required wave lengths. To this coil are connected the self-inductance coil of the antenna and the antenna itself. A variometer (*28*), consisting of three self-inductance coils, is provided for tuning the antenna to a particular wave length, and is likewise provided with a series of plug connections for this purpose. The apparatus is primarily arranged for transmitting the international wave length of 300, 450 and 600 metres, but is suitable for all lengths from 300 to 900 metres, and can by the addition of another coil transmit waves up to 1,600 metres. The operator can test whether the apparatus is in order by means of the hot-wire ammeter (*26*).

The receiving apparatus (*33*) consists of a variable plate condenser, transformer, detectors, and at the back a switch for the whole installation. By a simple turning of the condenser and adjustment of the transformer, any desired wave length between 200 and 2,500 metres can be obtained. For use in case of emergency, should the ship's electric engines not be running, there is provided a 500-watt accumulator battery (*44*) working an inductance coil (*43*). With this battery a range of 200 k.m. by day or 400 k.m. by night can be maintained for several hours. Communication between land and ship stations is generally such that the ship station transmits with a normal wave length of 600 metres and receives at about 2,000 metres wave length, as the land stations are not usually arranged to transmit the smaller lengths.

The transmission distance depends on the power of the installation and the height of the antenna. Fig. 2 shows the transmission distances by day for the various types of "Telefunken" apparatus as installed in different classes of ship, plotted in relation to the mean height of the antenna above the deck. At night the transmission distance is from two to three times as far.

THE ARRANGEMENT OF BOAT INSTALLATIONS ON MODERN SHIPS.

By Mr. A. Welin.

(*Institution of Naval Architects; March*, 1912.)

THE advent of passenger-carrying vessels of immense size has brought the authorities in most maritime countries to realize the necessity for revising the existing rules relating to life-saving appliances. New regulations have been formulated, and further important changes may be predicted in the near future.* These modifications by no means limit themselves to the number and cubic capacity of the boats, but refer also to the mode of installation of the boats and the efficiency of the launching gear. The object of the present paper is to describe some of the more recent boat installations on ocean-going steamers, and to indicate the possibilities of further development in efficiency.

An important consideration in connection with

* This paper, of course, was written prior to the loss of the *Titanic*. —*Editors.*

boat installations in passenger vessels is the deck space required for the boats. The larger this space the less is the promenade space available for the passengers. In this respect the double-

Fig. 1.—Welin Double-acting Davit.

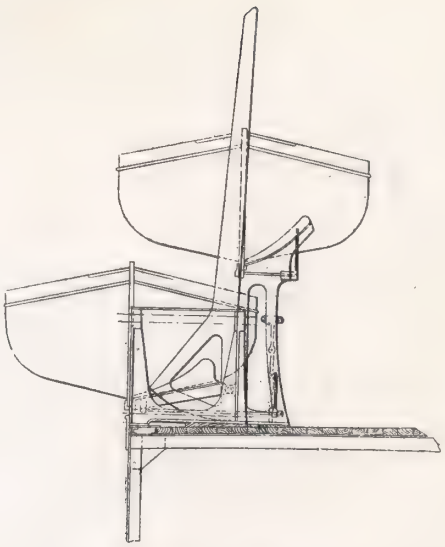


Fig. 2.—Arrangement of Boats for a Troop Ship.

acting type of davit devised by the author presents great advantages by making it possible to place the boats in double rows. With davits of the ordinary pattern it is not practicable to handle more than one row of boats. With Welin double-acting davits (Fig. 1) two rows can readily be handled, and when such davits are provided the Board of Trade permit one-fourth,

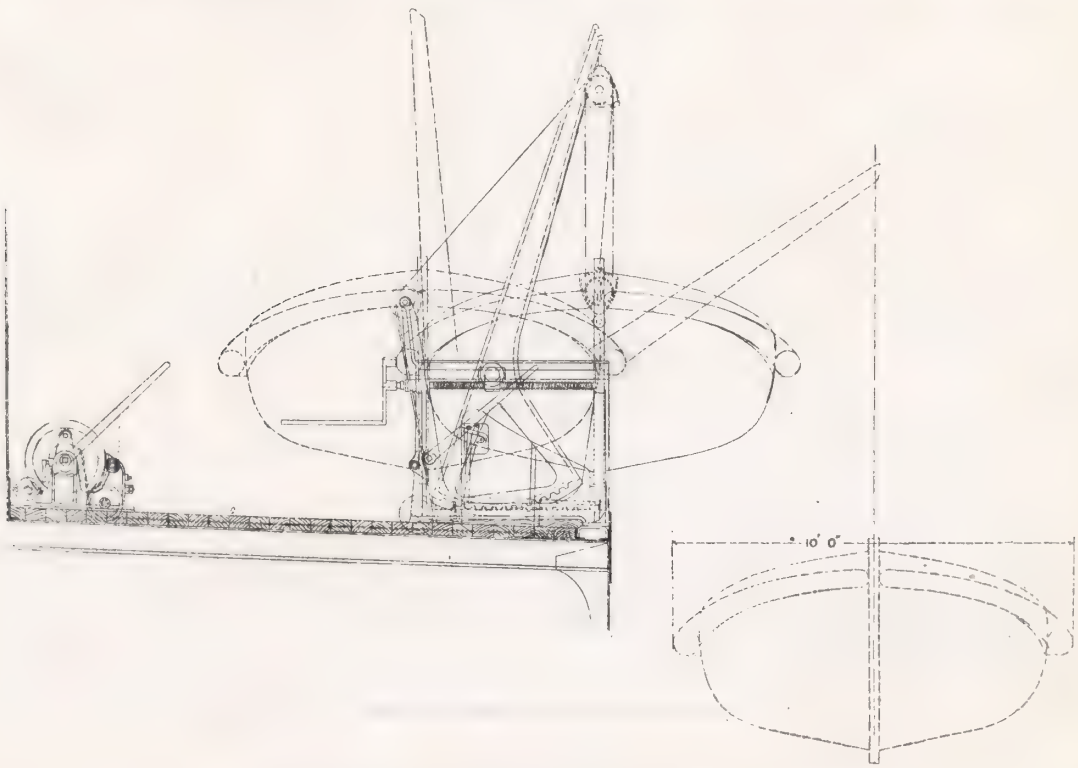


Fig. 3.—Welin Davits, Hamburg-Amerika Liner "Imperator."

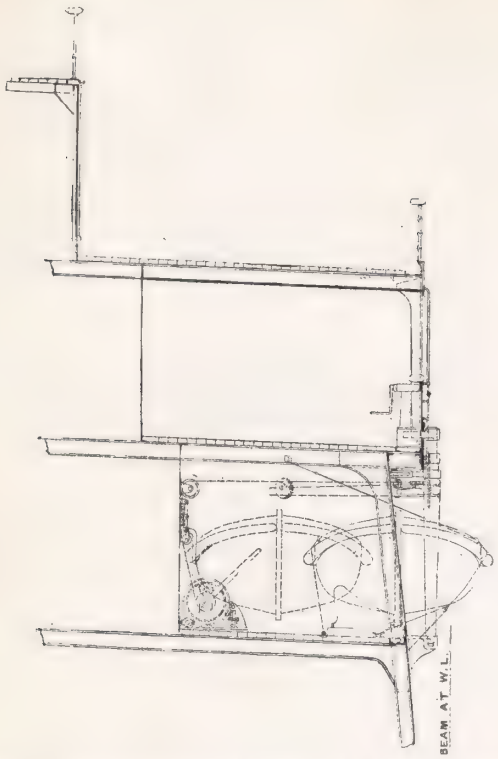


Fig. 4.—Section showing Boats on a Lower Deck.

and, where there are more than eight boats, one-third of the total number of boats to be placed inside the others and yet be counted as "standing under davits." In the case of the White Star liners *Olympic* and *Titanic* this double-acting type of davit was fitted throughout to make it possible to double, or even treble, the number of boats, should such increase ultimately prove to be necessary.

Exigencies of constructive design have naturally compelled the manufacture of many modifications of the Welin davit; yet it has been an invariable rule that the original simplicity should be maintained as far as possible. Fig. 2 shows an arrangement of boats specially designed for a troop ship. The lower of the two boats stands in this instance permanently half outboard. The davits are of the single-acting type with somewhat lengthened arms.

In another arrangement which is to be adopted for the new Hamburg-Amerika liner *Imperator* the davits are of the usual Welin type, but the boats are hoisted by means of a "control" (Fig. 3), which, when required, is thrown into gear by an electrically-driven fore-and-aft transmission shaft. Flexible steel wire falls are employed, and are run through shallow channels across the deck from the davits to the control. Requisite provision is made for any correction in

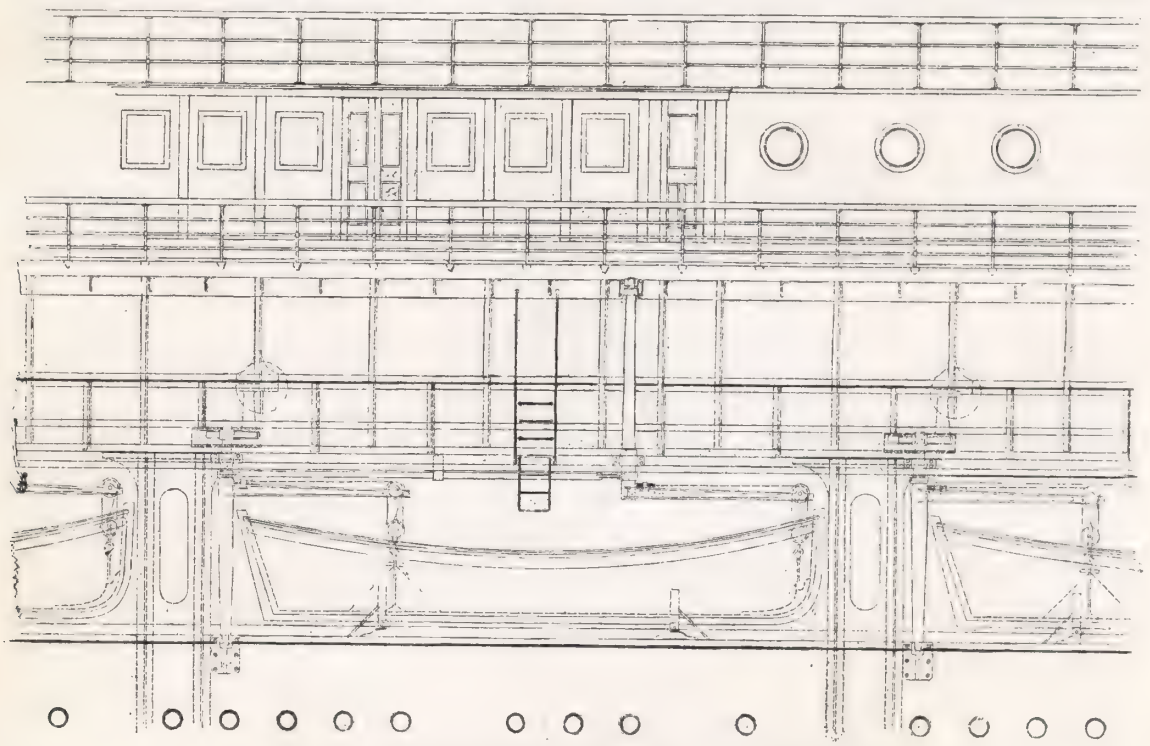


Fig. 5.—Elevation showing Boats on a Lower Deck.

the length of the falls, so that the launching can be carried out with the boats at any angle that the conditions of ship and sea may necessitate. Any mechanical contrivance which excludes such provision must, in the author's opinion, be regarded as dangerous and fundamentally wrong in principle. The launching manœuvre, which can be carried out in from 30 to 40 seconds, is entirely accomplished by hand, the power-transmission shaft being only required for hoisting. The power absorbed in hoisting the boats at an approximate speed of one foot per second is 10 H.P. per boat.

The author considers that in the largest steamers great advantages could be gained by removing the boats to a lower deck. The boats would then have a far better chance of reaching the water safely; the lowering would occupy less time; two decks would be available for filling the boats; unobstructed promenading space would be available on the top deck; and the stability of the ship would be improved. Such an arrangement is outlined in Figs. 4 and 5. The principal mechanical feature of the arrangement is that one at least of the two davits is supported *above* the boat. The two davits are connected by a coupling rod attached to short cranks on the davits proper; so that the arms stand at every point parallel to one another, the boat travels parallel with its own axis, and the tackles always remain in a vertical position. The turning-out gear is operated from the deck above. The actual constructional departures do not appear to be prohibitive.

In Fig. 5 the boats are carried on the shelter deck, recessed in bays spanning 36ft. centre to centre, and allowing clear openings of 32ft. 2in. The ordinary promenade deck supporting angles are dispensed with, and the strength is supplemented by extending two adjacent main frames of the vessel up to the deck above the boats, the space between these two frames being plated over.

In conclusion, the author draws attention to the advantage of combining boat-hoisting and coal-hoisting arrangements when it is possible to do so.

THE DESIGN AND MECHANICAL FEATURES OF THE CALIFORNIA GOLD DREDGE.

By Mr. Robert E. Cranston.

(*American Society of Mechanical Engineers; February*, 1912.)

DETAILED information in regard to all conditions is requisite in order to properly design a gold dredge. The first California dredge, built in 1897–1898, was patterned after the New Zealand type, and the present California type dredge is a combination of this type and several others.

The buckets are made with cast steel base, pressed steel hood and manganese steel lip. The lower tumbler is a six-sided steel casting with renewable wearing plates over which the buckets pass. They then travel up a structural steel ladder on rollers and over a six-sided upper tumbler which is driven by a chain of cast-steel gears. The buckets dump their material into a hopper which discharges into a shaking or revolving screen. The fine material goes through on to gold-saving tables and the coarse material is stacked aft of the dredge by means of a belt conveyor. The dredge is held in place by steel spuds and moved by means of side lines running to a motor-driven winch. The digging ladder is raised by means of a separate winch driven by the digging motor. Water is furnished under pressure by motor-driven centrifugal pumps. The hull is built of Douglas fir and should be designed last.

For rough estimating the cost of a modern California type dredge can be figured at about $180 per ton.

Fig. 1.—9-cubic-ft. Gold Dredge; 1911.

A typical 9-cubic-feet dredge is shown in Figs 1 and 2, to which the general particulars given in Table I. apply.

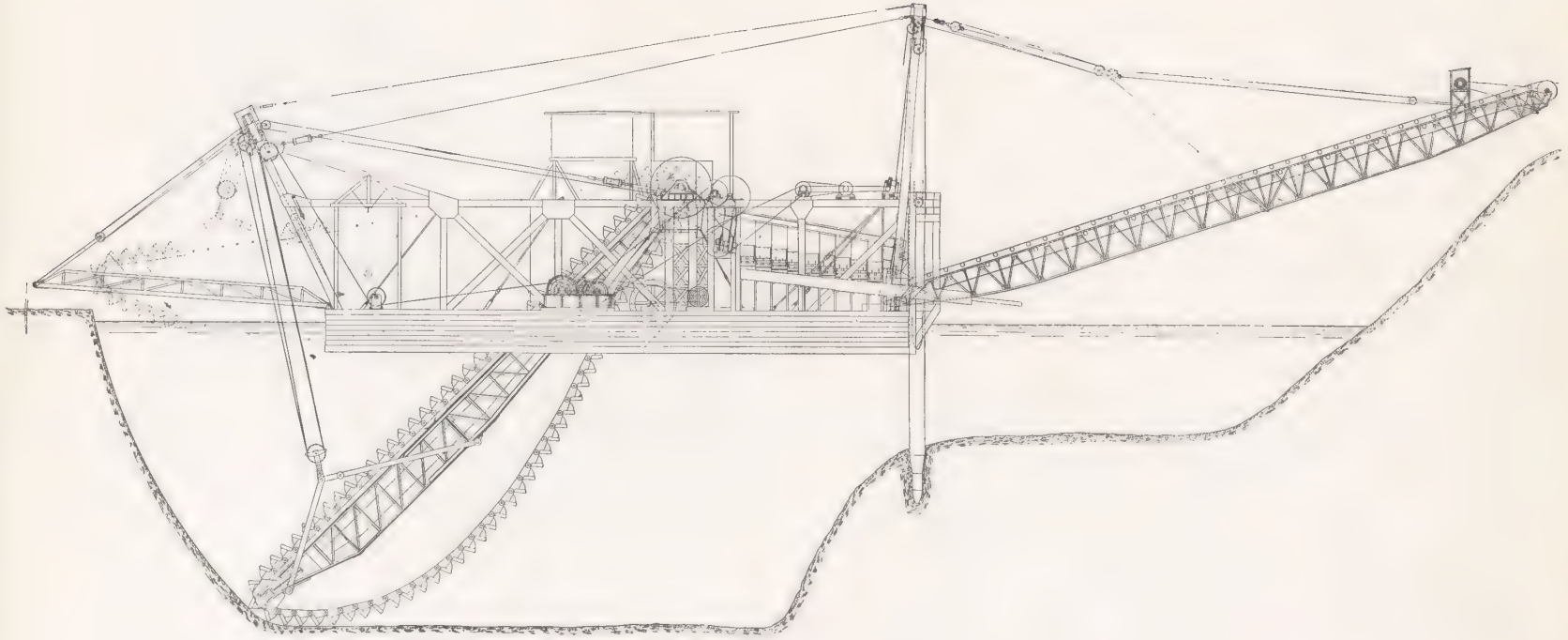


Fig. 2.—Elevation of 9-cubic-feet Gold Dredge; 1911.

Monsieur M. LAUBEUF,
Formerly Ingénieur en Chef de la Marine Française.

La navigation sous-marine est encore bien loin
d'avoir dit son dernier mot

M. Laubeuf

TABLE I.—GENERAL DATA OF 9-CUBIC-FEET. DREDGE.

Hull : total length, 115 ft.; width 50 ft.; depth, 8½ ft.; 200,000 bd. ft.
Digging ladder : total length, 115 ft.; maximum depth, 8 ft.; width, 4 ft. 7½ in.; weight complete, 94,400 lb.; built up of 5 in. × ⅝ in. L's and ½-in. plates.
Stacker ladder : total length, 140 ft.; depth, 7 ft.; width, 3 ft. 8 in.; weight complete, 22,500 lb.; built up of 4 in. × 4 in. × $\frac{7}{16}$ in. L's.
Bucket Line : eighty-four 9-cu.-ft. buckets; weight of bucket complete with pin and bushings, 2,200 lb.; pitch, 3 ft.; speed, 50 ft. per min.
Revolving screen : total length, 37 ft.; diameter, 6 ft.; weight complete, 37,360 lb.; built up of 6 sections, twenty-four ½-in. grizzly plates ⅜-in. holes.
Tables, sluices and save-all : total area, 2,430 sq. ft.
Spud : total length, 60 ft. 10 in.; weight complete, 36,000 lb.; built up of 4 in. × 4 in. × ⅝-in. L's and ⅝-in. plates.
Ladder winch : weight complete, 39,000 lb.; r.p.m. drumshaft, 7½; outside diameter, 3½ ft.
Main winch : weight complete, 23,275 lb.
Pumps : high-pressure, 2,800 gal. per min.; 50 ft. head; 10 in. discharge; 12 in. suction.
low-pressure, 4,000 gal. per min.; 25 ft. head; 12 in. discharge; 12 in. suction.
primer pump, 2-stage, 2 in. discharge; 2½ in. suction.
Motors : digging motor, 200 h.p.; screen motor, 50 h.p.; main winch motor, 25 h.p.; stacker motor, 35 h.p.; high-pressure motor, 100 h.p.; low-pressure motor, 35 h.p.; primer pump motor, 10 h.p.
Lines : ladder line 1⅛ in.–6/19–plough steel hoisting rope 1,035 ft. long.
spud line ⅝ in.–6/19–C. G. S. hoisting rope 553 ft. long.
stacker line ⅝ in.–6/19–C. G. S. hoisting rope 500 ft. long.
swinging line 1 in.–6/19–C. G. S. plough steel hoisting rope 600 ft. long.

TORPEDO BOATS AND TORPEDO-BOAT DESTROYERS.*

By Monsieur M. LAUBEUF.

(*Société des Ingenieurs Civils de France ; May*, 1912.)

THE invention and improvement of the torpedo led to the construction of a large number of vessels specially designed to utilize the new weapon. These vessels, known as torpedo boats, were small in size, light in construction, and possessed propelling machinery of sufficient power to give them a considerable speed. By 1895 the type had developed to a length of 150ft., a displacement of 180 tons, and a speed of 28 to 30 knots. The *Forban*, completed about this time by the celebrated designer and builder of torpedo boats, M. Normand, realized a speed of 31 knots, with about 3,500 I.H.P., the displacement being only 130 tons.

This type eventually became too small to accompany fleets, and has been abandoned in favour of the larger type of vessels known as torpedo-boat destroyers. The torpedo-boat destroyer is not merely a vessel for firing torpedoes. Such vessels also possess an armament of guns, and the factor of gun armament is beginning to take precedence over torpedo attack in the design. The name torpedo-boat destroyer is really a misnomer. A more appropriate name, in view of their functions, would be despatch boats of the fleet. The French naval authorities call them *torpilleurs de haute mer* (sea-going torpedo boats). M. Laubeuf adopts the name *torpilleurs d'escadre* (torpedo boats of the fleet).

The first French destroyer, the *Durandel*, was built at Havre to the designs of M. Normand in 1896. This vessel had a length of 181ft. 6in., a displacement of 300 tons, and obtained a speed of 27 knots with twin propellers each driven by a triple-expansion engine. She was armed with one 65-mm. gun, six 47-mm. guns, and two torpedo tubes. The size was slightly increased in the *Claymore* class which followed, the displacement becoming 337 tons. Altogether fifty-five vessels were built of the two types, two of which achieved a speed of 31 knots.

In course of time the dimensions again proved insufficient, as in rough weather these destroyers could not keep up with the newer battleships. In 1906, therefore, the construction was commenced of vessels having a length of 215 feet, a displacement of 450 tons, and an armament of six 65-mm. guns and three torpedo tubes. Thirty of this class were built between 1906 and 1911.

The vessels of 450 tons displacement may be divided into three classes, in which the propelling machinery consisted respectively of reciprocating engines and twin screws, turbines and triple screws, and a combined arrangement of one reciprocating engine and two turbines. About the middle of this series a commencement was made with the use of oil fuel for firing the boilers. Two vessels of this class, the *Voltigeur* and the *Tirailleur*, deserve special mention as being the first destroyers in which turbines of French design were installed. The *Voltigeur* was fitted with Rateau turbines in combination with a

* This paper arrived too late to be included in the first section of the volume.—Editors.

reciprocating engine, and attained the highest speed of any of the vessels belonging to this class, *viz.*, 31·3 knots. The *Tirailleur* was fitted with Breguet turbines and a triple-expansion engine.

In 1908 it was found desirable to still further increase the size, and a new class of destroyer was ordered having a displacement of about 700 tons. The increased sea-keeping qualities of this class were attained not only by the increase of size but also by the provision of a forecastle. The speed asked for was 31 knots, but this has been exceeded by many of the vessels, notably the *Bouclier* constructed by Normand and Co., which attained a speed of 35·34 knots on the six hours trial, and the *Casque* built by the Société des Forges et Chantiers de la Méditerranée, which attained 34·9 knots. As armament this class have two 100-mm. guns, four 65-mm. guns, and four torpedo tubes placed in pairs, from which it will be seen that the predominance of gun over torpedo armament has been carried a stage further.

It appears to the author that the increase in size, while providing an efficient type of scouts for the fleet, has resulted in a sacrifice of the qualities desirable for torpedo-firing, *viz.*, invisibility and possibility of quick manœuvring into a suitable position for firing, which are associated with a small vessel. He considers, therefore, that two types of vessel should be built, one having a displacement of about 1,000 tons and embodying all the qualities required in an efficient sea-going scout, and the other, intended for firing torpedoes only, having a displacement of 200 to 250 tons and a speed of 28 knots. This idea of two types of torpedo craft was for a time adopted in Great Britain, and thirty-six so-called coastal destroyers, in effect torpedo boats, of from 250 to 300 tons displacement, were built for the British Navy between the years 1906 and 1909. The same principle has been adopted by the Austrian naval authorities.

With regard to propelling machinery, M. Laubeuf considers that turbines are undoubtedly preferable to reciprocating engines. At the same time the turbine possesses the disadvantage of a large steam consumption at the reduced speeds which warships usually adopt for cruising.

In respect to different arrangements of turbines, the author points out that the twin-screw arrangement, with an independent unit driving each shaft, possesses important military advantages, and also leads to a simple pipe system. In this case, when proceeding at low speeds, one of the units can be laid idle. Under such circumstances the following interesting figures were obtained on the destroyer *Fourche* :—

	With one turbine.	With two turbines.
Duration of trial	29 hrs. 10 m.	15 hrs. 5 m.
Speed	12·89 kts.	12·9 kts.
Consumption per hour per knot	89 kg.	96·5 kg.

A rudder angle of 3 deg. sufficed to keep the vessel on her course when proceeding with one screw alone.

The three-turbine Parsons arrangement leads to a smaller fuel consumption, but entails much greater complications in the installation, and the author is of opinion that for military reasons the two-turbine system is preferable, in spite of the reduced economy. The large number of twin-screw destroyers building for various navies confirms this view.

M. Laubeuf then gives a list of the destroyers built and building for the British, German, Austrian, Italian, United States, Russian, and Japanese navies. In conclusion, he points out the grave deficiency of the French Navy in such craft. The French possess only 18 vessels of more than 600 tons displacement, while the British have 108 and the Germans 72. The naval programme provides that 52 sea-going destroyers shall be in service by 1919, which means that 34 have to be built before that date. The budget for 1913 only includes provision for two, which is quite insufficient if the programme is to be fulfilled. The vessels should all be ordered in the four years 1913 to 1916 if they are to be ready by 1919. They should have a displacement of at least 900 tons and preferably 1,000 to 1,100 tons, a speed of 33 knots, and an armament of six 100-mm. guns and four torpedo tubes. Vessels of smaller displacement than 600 tons should be transformed into torpedo boats pure and simple, for coast defence. Their radius of action should be reduced and the number of torpedo tubes increased, four torpedo tubes being fitted instead of two in vessels of 300 tons and five instead of three in vessels of 450 tons displacement.

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